Design of an Impact Ram Used to Simulate Loadings Encountered During Automobile Accidents

A Thesis

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Abstract

In tests used to assess the impact biofidelity of anthropomorphic test dummies (ATD), post mortem human specimens (PMHS) are impacted with a ram to simulate the loadings that occur during automobile impacts. Other tests require the ram to impact the ATD for calibration and verification. The current pneumatic impact ram in the Injury Biomechanics Research Lab (IBRL) has become obsolete primarily because of the inability to accurately control the velocity of the ram as it impacts the subject. Because of the inaccurate and unpredictable nature of the current ram, redundant and expensive tests may be required on an already limited supply of PMHS. Other issues have also become prevalent such as the long changeover period of the ram face and mass when different tests are to be conducted on the same subject, consuming precious time by allowing the cadavers to further degrade. In other cases, the angle of the camera equipment must be adjusted relative to the ram during impacts at an angle when ideally the ram would adjust relative to the camera equipment because of the confined space. It is the objective of this research project to:

- Determine design requirements of a new ram to achieve impacts at a constant velocity given varying impact ram mass and velocity requirements of current ATD and PMHS tests [1]
- Determine the instrumentation to measure the displacement, velocity, acceleration, and force of the ram, and to consider the possibility of adjusting the height, the angle of impact, and the mass/face of the ram
- Design and document a new impactor to meet the agreed upon design requirements

I would like to dedicate this dissertation to my fiancée, Debbi, and to my family and friends who have supported me throughout my undergraduate career.

Acknowledgments

I would first like to acknowledge my advisor Dr. John H. Bolte IV for providing me with the opportunity to work in the Injury Biomechanics Research Lab and for his guidance throughout the course of this project. My time spent with Dr. Bolte has helped me develop a bridge between mechanical engineering and medicine, and has reaffirmed my decision to attend medical school next autumn. In addition, he has always been there for support whether it was in regards to the project or personal issues outside of the project.

Next, I would like to acknowledge Rod Herriot and Jim Clevenger for their help with engineering design. They made themselves available for discussion about various aspects of the design specifically in regards to feasibility, and have helped me consider other design ideas that I would not have considered otherwise.

Finally, I would like to acknowledge Professor Necip Berme for serving as a faculty examiner for my thesis defense and Jim Schmiedeler for helping me with my presentation skills with this project. I believe that in general, faculty member accessibility is a great asset for the Department of Mechanical Engineering at The Ohio State University.

Vita

September 15, 1985…………………………………………………………………………Born- Coldwater, Ohio

Fields of Study

Major Field: Mechanical Engineering

Minor Field: Life Sciences

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Introduction

There are several variations of impact rams used to develop and calibrate crash test dummies. The most common design has been the swinging pendulum because of the essentially nonexistent frictional losses in the system and the consistent final velocity due to its simple design. However this is not an option in the IBRL because of the physical constraints of the building. The height of the ceiling in the lab is much too low, and moderate impact velocities and stroke lengths could not be obtained. The best option for the IBRL is a hydraulic or pneumatic linear impact ram similar to the current design that will allow testing in a much more confined space. These rams can offer reliability comparable to the pendulum rams with the added flexibility to impact constrained objects that are sometimes out of reach of the swinging pendulum and its cables. The downsides to these systems include the increased complexity and troubleshooting due to mechanical wear and friction. The availability of these systems is also very limited because of the high force and velocity requirements, and the confined space of the lab. Off the shelf systems are very expensive and typically require a large lab area with reinforced flooring.

Background

Current System

The current ram features a piston cylinder assembly using compressed nitrogen as the energy source. The nitrogen is released into an accumulator tank until the pressure reaches a predetermined level. Once the tank is filled, a quick release valve is opened venting the gas into a cylinder, forcing a plunger forward until vents on the sides of the cylinder are reached, at which point the plunger decelerates, and the ram continues toward the object with a theoretical constant velocity. The piston cylinder assembly can be seen in Figure 1 with a spare piston positioned on the side of the ram. The venting location can also be seen on one side of the cylinder.

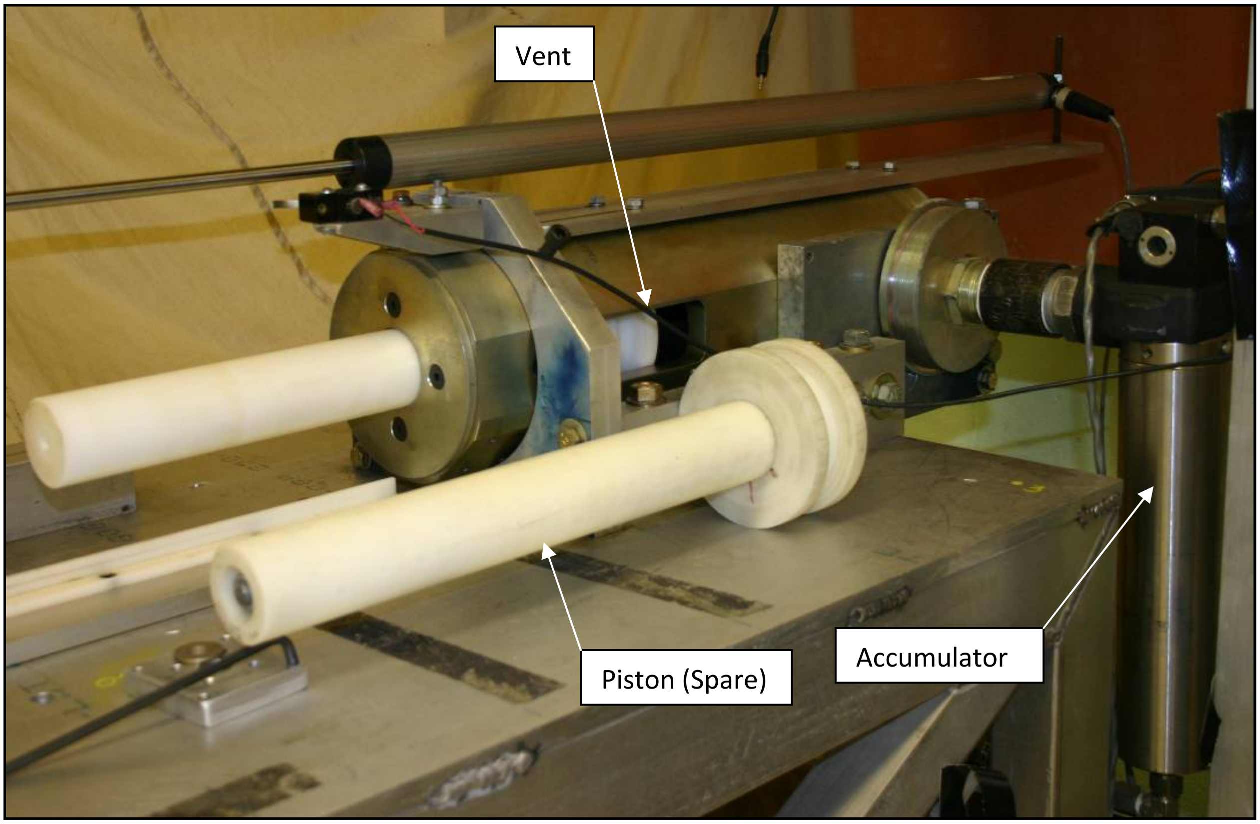


Figure 1: Current Pneumatic Accelerator

Motivation

The current system has many design flaws that cause complications on test days, the most detrimental of these being an irregular pressure velocity relationship. Figure 2 is a plot of the pressure velocity relationship for the current ram. This plot demonstrates the erratic behavior of the ram in the last year with a large deviation in the data points, especially at low velocities where the velocity of the ram grows even more unpredictable due to the friction of the nylon linear bushings.

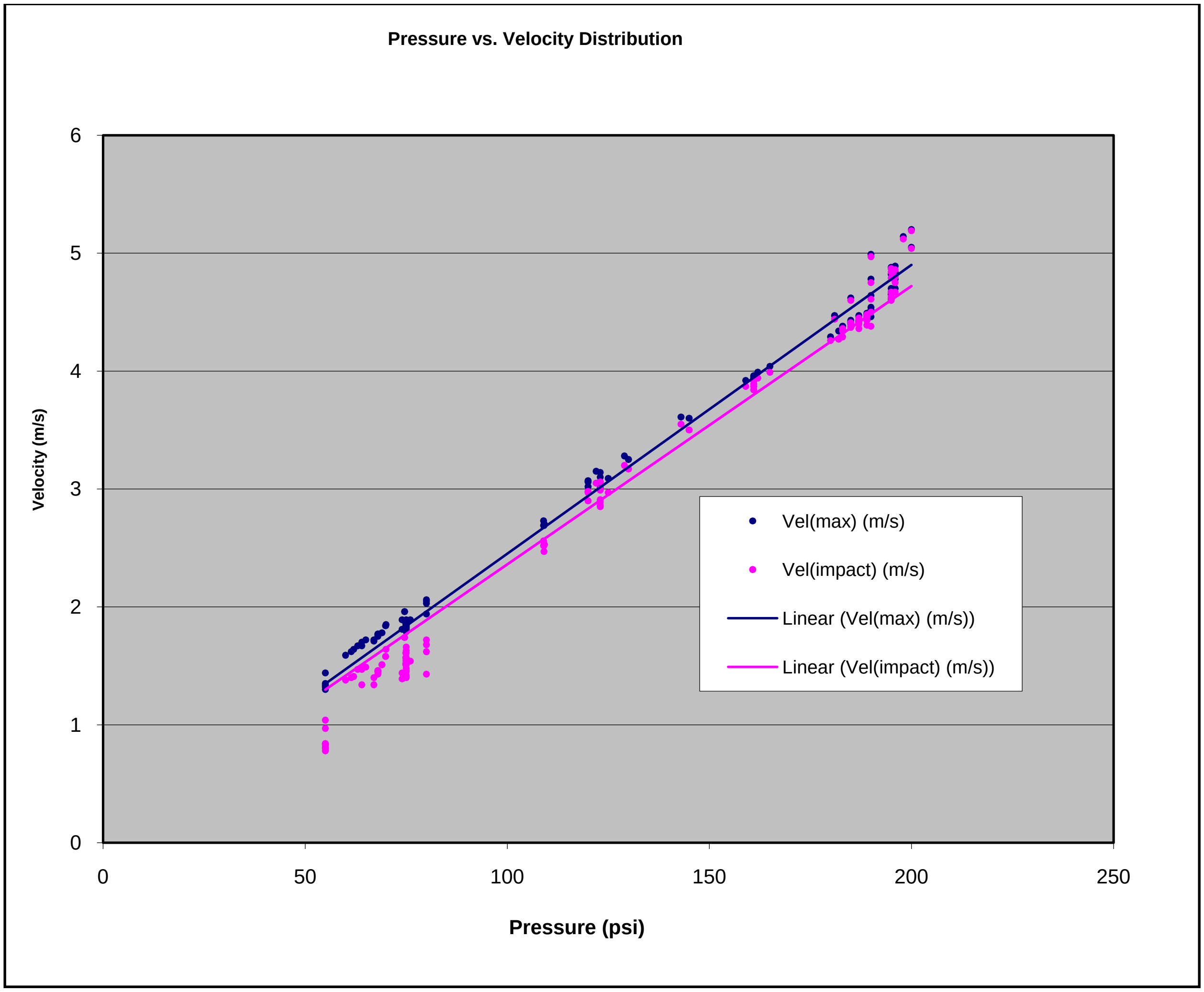


Figure 2: Pressure Velocity Variation

Altering the ram impact face and/or weight can consume 1-2 hours on days where lab personnel are spending upwards of 12 hours in the lab. These changeover periods are unnecessarily long, and can affect the data if the cadavers are required to sit in a warm setting for too long. A few of the different faces and weights can be seen in Figure 3. The load cell (not pictured) mounts between the ram weight and face of the ram. To change the weight of the ram, the entire ram assembly must be disassembled which is a very difficult procedure because the shaft can freely rotate making it difficult to loosen and tighten bolts. The confined spaces between the weights and faces make the process much more cumbersome and time consuming. Finally, when the ram is reassembled, the behavior can be unpredictable because of the different moments placed on the bearings (nylon bearings have higher static friction).

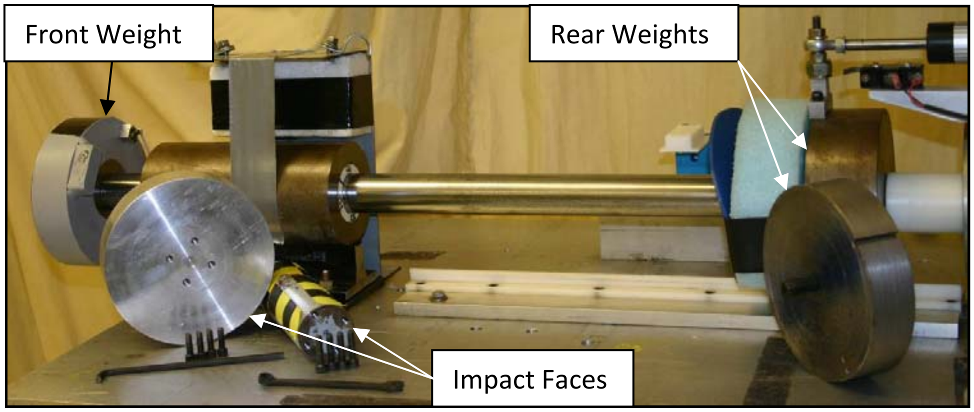


Figure 3: Impact Ram Weights/Faces

Other shortcomings of the ram include the inability to adjust the ram height or firing angle. This limits the ram in cases where the heights of test subjects are not average and in tests where the ram needs to be fired at an angle to the subject. Having the ability to rotate the ram instead of the subjects will allow for a uniform data collection procedure because lights and video cameras will be able to be placed in the same location for most tests. Also, adjusting test subjects is very tedious, and having the ability to make the adjustments on the ram would allow for better data collection.

Literature Review

Because this project was very design oriented, the literature review was typically limited to vendor catalogs and specification sheets. Vendor information for various aspects of ram design can be found in the Appendix. Additional resources are clarified when necessary.

Overview of the Design

To most clearly outline the selection process for various aspects of the ram design, the Design Specifications and Constraints section will first orient the reader with specific issues and concerns regarding ram design and will clarify the goals for the new impact ram. The Proposed Solutions section will then consider several design options for the accelerator and analyze why each option was or was not chosen. Next the final design will be discussed in detail, and the budget and other considerations will be considered. To conclude, future concerns will be discussed along with a conclusion detailing which goals have been reached.

Design Specifications and Constraints

A list of the most important specifications for the impact ram has been developed. Each of the specifications is further detailed in the follow sections.

- Achieve a speed of 10 m/s with a mass of 23.5 kg
- Achieve an accuracy of ±0.05 m/s
- Negligible & predictable friction
- Low noise
- Impact ram axially constrained
- Minimum mass of 10 kg
- Inertially compensated load cell
- Rigid & resilient support structure
- Size and weight of individual components
- Compatible with drop tower/overhead support system
- Vertical adjustment between 18" and 48"
- Angular adjustment ±15°

Accelerator

The design specifications for the accelerator circuit first and foremost include the requirement to achieve a ram speed of 10 m/s with a mass of 23.5 kg with and a maximum acceleration length of 0.4 m though the goal length is 0.3 m. These specifications were the maximum foreseeable values based on current test requirements [1] and half of the weight of a 50th percentile male torso. The new kinematic specifications are detailed in Table 1 relative to the current system. The inertial forces for these velocities assuming constant acceleration are shown in Figure 4 and Figure 5 below. Figure 4 demonstrates the forces at the maximum acceleration length of 0.4 m, while Figure 5 displays the required forces for the goal acceleration length of 0.3 m. It is also important to note that in these figures, the forces were calculated assuming no smoothing (constant acceleration) which would not be the case if the motion profile were to be implemented and zero friction forces when in reality, the ram may experience friction forces from a less than perfect linear bearing.

Table 1: Accelerator Kinematic Specifications

	Current	Goal	Max
$Velocity_{max}$ (17 kg old, 23 kg new) m/s	10	10	10
$Mass_{max}$ (7.5 m/s) kg	23	37.5	37.5
Acceleration Length m	0.23	0.3	0.4
Deceleration Length (Considering $Velocity_{max}$) m	0.03175	0.0762	N/A

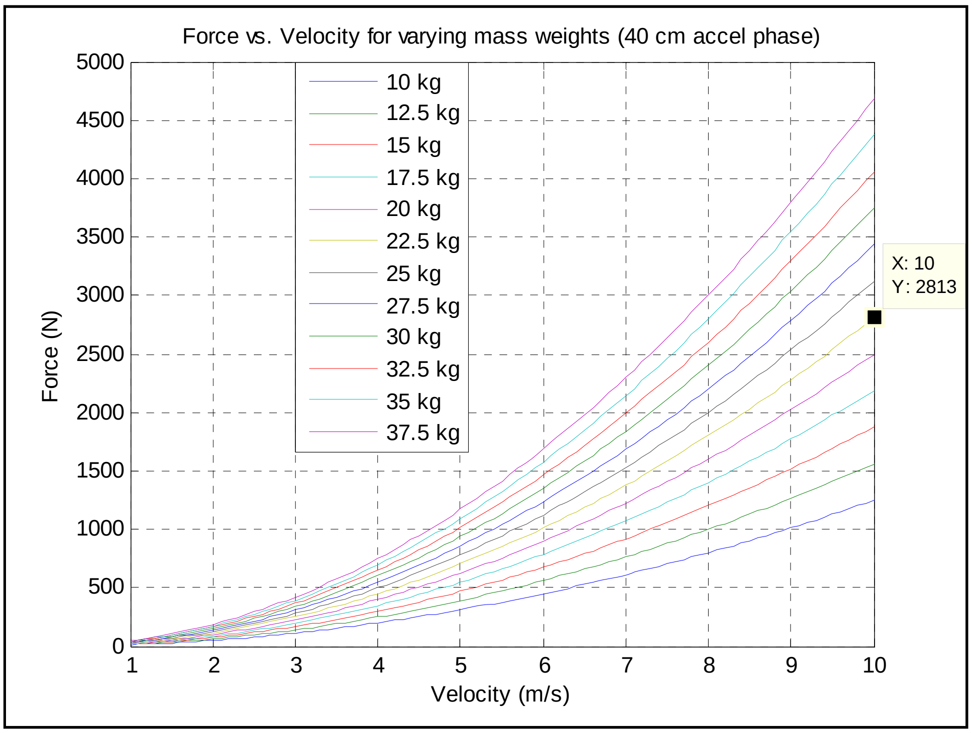


Figure 4: Inertial Force Requirements Assuming 40 cm Acceleration Phase

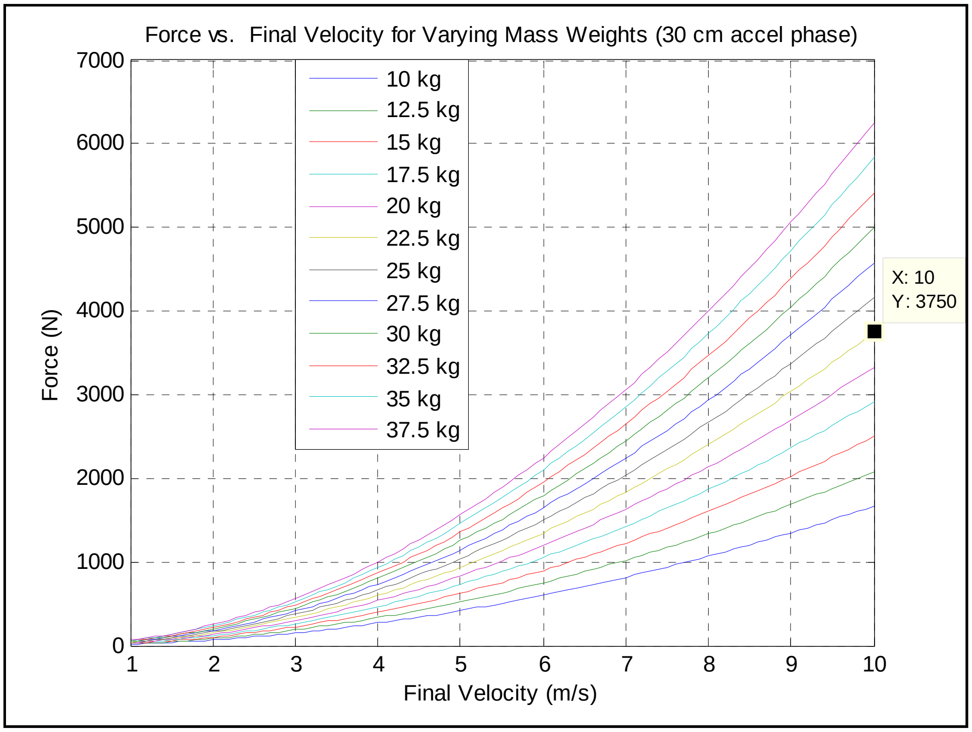


Figure 5: Inertial Force Requirements Assuming 30 cm Acceleration Phase

One aspect of the above figures that will be discussed in some detail later in the design constraints of the support structure is the relatively low force requirements because of the constant force applied to the ram. The current pneumatic ram experiences a much higher peak force because the discharging of the cylinder acts much like a firearm given that almost all of the acceleration happens within the first few moments after the valve is opened.

Another independent constraint for the accelerator would be the requirement for an energy system that would be relatively mobile to complement the movement of the ram in the vertical direction (and rotation about the vertical axis).

Control System

The requirements for the control system include the ability to accurately achieve a final velocity of the ram within a maximum of ±0.05 m/s and to maintain this velocity for a short duration of time. The accelerator will then be required to either remove itself from the path of the ram should it return, or be capable of sustaining a reverse impact and decelerate the ram to some degree.

The control system must have a simple user interface, and would ideally include the ability to adjust the acceleration length based on the testing requirements. For instance, there may be instances when using a heavy ram where the entire acceleration distance would be utilized and other instances when using a less massive ram where the acceleration length would be shortened in order to provide more room to accommodate the test setup. The lengths for the various phases of the acceleration and impact of the ram that the control system must match are outlined in Table 2.

Table 2: Impact Ram Travel Lengths (Co-linear to Ram Travel)

	Current	Goal	Max
Accel Length m	0.23	0.30	0.40
Decel Length (Considering Velocity$_{max}$) m	0.03	0.08	0.08
Coast Length (B/t Accel & Stroke) m	0.06	0.03	0.08
Stroke Length (After Contact) m	0.20	0.30	N/A
Overall Structure Length m	2.51	2.51	2.79

Impact Ram

Bearing System

There are a limited number of requirements specific to the bearing system for the impact ram. The first is that the impact ram should have very low friction to enable minimum velocity loss from the time the accelerator begins to lose contact with the ram, to the point of impact. Because there will inevitably be a nominal amount of friction in the system, the friction must at minimum behave predictably so that the velocity loss can be accurately estimated to obtain the desired velocity at impact. This will mean that the friction should not vary with velocity, and that the static friction should approximate the kinetic friction.

The next design specification requires that the bearing system produces very little noise because introducing noise to the system can disrupt the function of the accelerometers and other data collection devices. This requirement will potentially tie in very closely to the first because a system with low friction will often include high noise. The most obvious choice for a low friction system would feature ball bearings which would create much high frequency noise. If ball bearings are to be used, they must either have very little noise, or a different data collection device resistant to high frequency noise must be employed.

The final design requirement for the bearing system is that the impact ram must not have free rotation. This will allow for an easier ram face/mass changeover, and will help prevent sensor wires from wrapping around the ram or getting caught on the frame.

There are no constraints for the bearing system aside from those mentioned in the specification section.

Face Changeover

The specifications for the face changeover include the requirement for the ram to be capable of accepting several different face and weight configurations between 10 and 37.5 kg. Although the maximum kinematic requirement calls for 23.5 kg to be accelerated to 10 m/s, it may still be desirable to run tests utilizing the weight of 37.5 kg at lower speeds up to 7.5 m/s. Initially a plan utilizing mass increments of 2.5 kg was chosen. However, this setup may be a disadvantage because test requirements and face configurations may not always specify these discrete masses. A better solution might involve incremental weights for the rear of the ram for balance to get close to the desired total mass. The final total desired mass would then be achieved by controlling the precise weight of the impact ram face to the particular value. This would allow for a more customizable system where the ram could utilize an infinite number of masses within the range of 10-37.5 kg.

The single greatest constraint for the mass/face changeover of the ram is anticipated to be the vibration/noise that will be created if the weights/faces are mounted securely. Using a plate system for the mass changeover may introduce a problem because of issues associated with the plates crashing together on impact. The current ram uses a series of axially mounted hex screws to tighten the faces down, which works well except that the rod has free rotation and tends to spin when trying to remove the hex nuts. If the ram were to have a groove or hole that could be located to fix the rotation of the ram, this problem would be resolved.

Instrumentation

The design specifications for the instrumentation will not be considerably revised from the current design. The one potential requirement as mentioned before would the signal conditioning for the accelerometers in the case where low noise bearings could not be purchased. The kinematic values that need to be measured include displacement, velocity, acceleration, and force. This could be done with a series of linear displacement potentiometers, accelerometers, and load cells. The load cell would preferably have inertial compensation, and would be a six axis load cell so that all moments and forces could be calculated. If necessary, a speed trap would also be utilized.

Support Structure

Test Frame Assembly

The test frame assembly must support the ram during impact testing, and help to absorb the shock as much as possible to prevent forces from being transmitted to the building. Because the frame must be rigid for purposes of data collection, the best way to absorb the shock will likely be with a shock absorbing pad placed on the floor beneath the supports. To prevent movement of the support structure during testing, the ram will be situated against the back wall, and is expected to utilize seismic buffers to dissipate the high frequency vibrations before they enter the floor. Actual values for the maximum force entering the building are not anticipated to be significant relative to building structure.

The test frame has several constraints imposed on it by the building size with respect to overall mass and dimensions. The lab is located on the 3rd floor of Graves Hall, and the load capacity of the freight elevator is rated at 4500 lbs, so no individual component can exceed that weight. Dimensionally the impact test frame has potentially the most challenging constraints regarding the size of the individual components. The dimensions of the door entering the lab are 42.75” wide by 79.25” tall. The dimensions of the freight elevator doors are 42” wide and 83.75” tall, and the freight elevator is 105” deep with a width of 56.75” and a height of 86”. These dimensions may require that the test setup use mechanical fasteners in some places that would be fastened in the lab where welding may have been desirable.

Another constraint for the test frame assembly is that the main vertical support beams to which the guide blocks are to be mounted must be parallel within a very tight tolerance to allow smooth vertical adjustment of the ram. This may require that one of the vertical support columns be slightly adjustable if the tolerances are not capable of being met. Additionally, the test frame assembly must be compatible with the current drop tower/overhead support system that is currently in use in the IBRL lab. This may be as simple as a width requirement, or as complicated as an entirely new overhead system that will attach to or straddle the new support structure.

Vertical/Angular Adjustment

Ideally, the impact ram would have a variable height between 18” and 48” measured from the center of the ram axis to the lab floor. The height could be adjusted using one or two power screws and a series of eight guide blocks for stability. The screws should be powered with a hand crank or a drive shaft that will be coupled to a small motor. If the motor set-up will be used, two gearboxes will have to be purchased to get a significant gear reduction with a 90° bend. A worm/spur gear combination may be appropriate for this because it will not be back-driven. If a hand winch is to be utilized, hand brakes will need to be added to the linear guides to prevent vertical movement. It is likely that this will be the more cost effective method.

The guide blocks must travel freely and the linear guides must be able to withstand the high forces associated with the firing of the ram. The maximum force experience by the test frame was estimated conservatively to be about 35,500 N based on the current ram system. This value was estimated using two different methods with the larger value selected for safety. This larger value represents the absolute maximum force because the calculation was based on the maximum accumulator pressure of 500 psi. It was assumed that when the valve is opened, the back side of the cylinder (ø 4.5") will experience 500 psi. In reality, the cylinder will not endure the entire pressure force because the ram will begin to accelerate immediately before the pressure can build up. However, this value will yield a conservative estimate of the total force. The less conservative value was determined by analyzing the kinematic data from the data acquisition system to determine the maximum acceleration. Even though the duration of the force lasted far less than one hundredth of a second, the maximum acceleration for a high speed test appeared to be about 80 g's. When this was converted to force assuming a worst case scenario of a 23.5 kg mass, the maximum force becomes 18,500 N; significantly lower than the conservative estimate based on the accumulator pressure. Because of the impact nature of these forces and the uncertainty in measurements, a minimum safety factor of 2.0 will be considered. (Note that these forces are significantly higher than those estimated in Figure 4 and Figure 5 because of the non-linear acceleration).

Additional forces that must be absorbed by the linear guide include rotational forces that be introduced because of the offset weight of the ram after the shot is fired. Attempts to balance the carriage assembly will be made, but during the firing of the ram it will be impossible to balance the weight. All of the above mentioned dynamic forces and moments are outline in Figure 6. This figure uses a previous design for the test frame because it more clearly depicts the members (because of the more open design) while providing a very similar example of the final design.

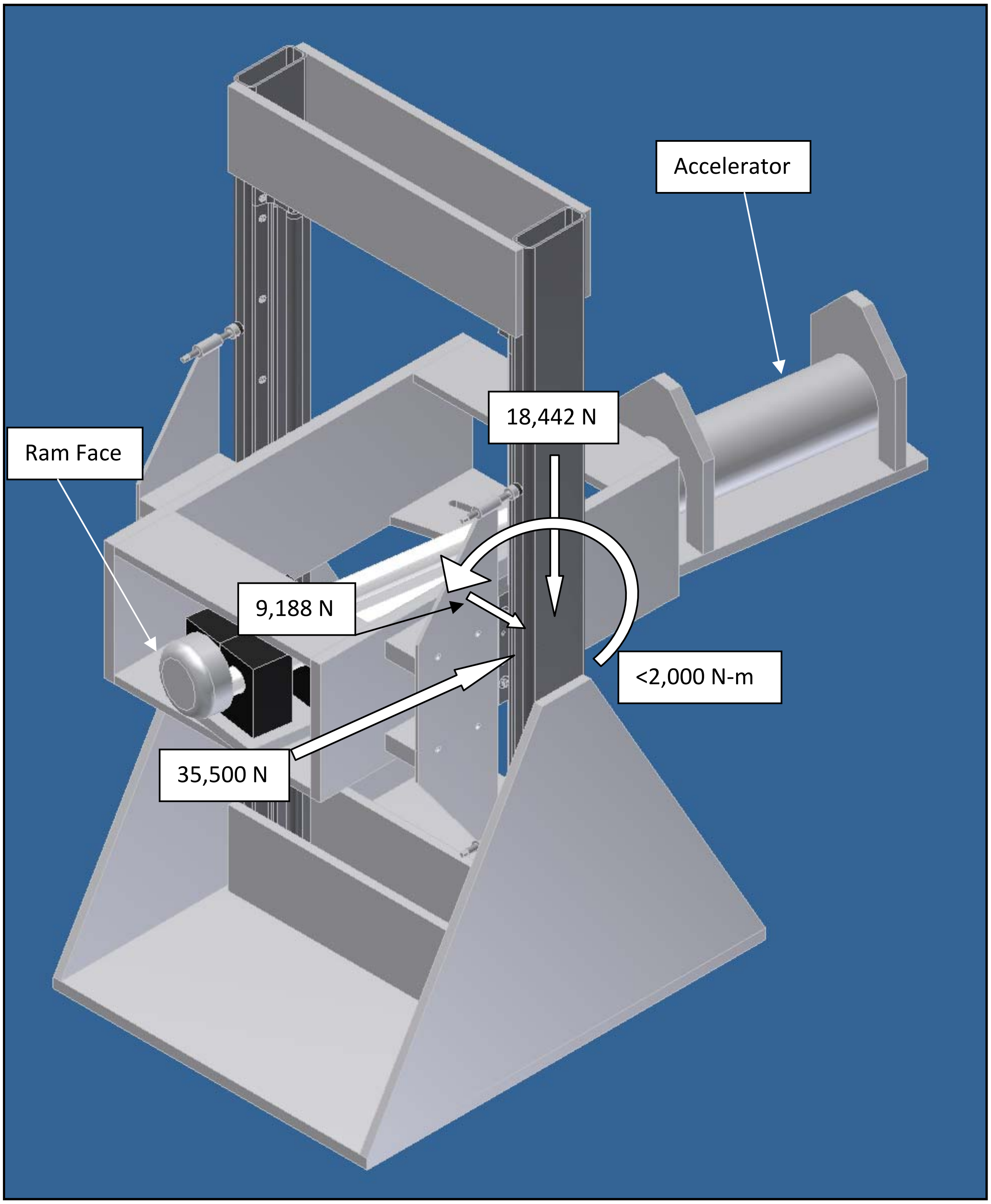


Figure 6: Preliminary Design to Demonstrate Maximum Forces

The other requirement for the support structure is that the impact ram be able to rotate about the vertical axis ±15°. This will allow the subjects to be adjusted relative to the testing equipment including cameras and lights. The mounting location should minimize the sweep area of the ram so that the subject will not have to move excessively in the plane perpendicular to the firing axis of the ram. Figure 7 demonstrates the smaller sweep area that can be realized by mounting the pivot of the horizontal beam near the test subject as opposed to the rear of the structure.

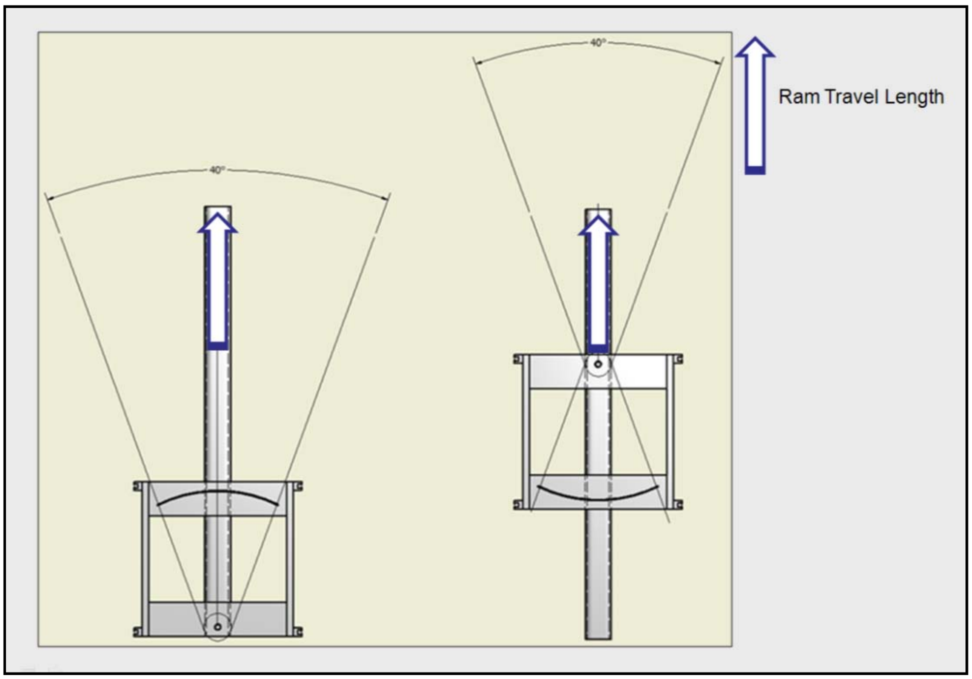


Figure 7: Approximate Relative Sweep Area of Impact Ram, Top View

Proposed Accelerator Solutions

VRTC Hydraulic System

The initial accelerator that was explored, a hydraulic system was found as part of an impact ram at the Vehicle Research & Test Center in East Liberty, Ohio. This ram was found to have improved operating characteristics relative to the current ram, but also shared many design flaws including slight leaks in the pressure lines and a cumbersome firing procedure.

Because this was a custom system, a simple capability study of the impactor was done and compared to the current pneumatic ram system. Limited documentation on the design and construction of the two rams was found, but considerable information was obtained from the Linear Impactor Brake Evaluation [2] for the hydraulic ram. To extend the evaluation a little further, speed tests were conducted to learn more about the pressure velocity characteristics of the two impactors (hydraulic from VRTC, pneumatic from IBRL). Finally, the pros and cons of the hydraulic ram were considered relative to the current pneumatic impactor

Purpose:

Pressure vs. velocity data was obtained to determine the operating characteristics of the linear impactors, primarily with respect to repeatability. Five speed shots were fired at five predetermined pressures within the normal operating range of the ram in the IBRL. It was the goal to fire five shots within the same velocity range with the hydraulic ram at VRTC, but it was soon discovered that the normal operating velocities for the hydraulic ram where above those of the pneumatic ram, so speeds shots at lower velocities may not have been a good indication of the capabilities of a similar hydraulic ram designed for lower speeds.

Setup:

The five speed shots from pneumatic impactor were taken using a ram weight of 23.67 kg with the wooden face designed for speed shot testing. The ram was decelerated using sandbags approximately ten inches from the firing position to assure that the ram could reach a maximum velocity. The pressure was measured using a pressure transducer in the accumulator tank, and velocity was calculated by integrating data from an accelerometer.

The ram weight for the hydraulic impactor was 30.75 kg during speed shot testing, and the ram was again decelerated with sandbags. Velocity and acceleration data was obtained with an accelerometer and a speed trap at 127 mm. Pressure was measured with a pressure transducer, though the pressure data appeared to be of too low resolution for a precise analysis.

Results:

From the pneumatic ram data, two different sources of inaccuracies become apparent when looking at the data. At low velocities, deviation occurs during the stroke length (constant velocity), as opposed to higher velocities where deviation appears to be a result of inaccuracies in the pressure velocity relationship. There was also a significant source of inaccuracy due to operator error because firing the ram at a predetermined pressure was difficult.

The results for the hydraulic data were less beneficial because a leak in the firing mechanism led to inconsistent firing conditions. However, one potential benefit of a hydraulic ram was that a large apparent deviation in pressure corresponds to a small deviation in velocity. Though only three speed shots were fired at four different pressures, the standard deviation for the velocities was never more than .02 m/s. Implementing a firing mechanism that would engage at a predetermined pressure could lessen operator inaccuracies in both cases.

Tabulated results and figures can be found in Appendix A.

Conclusion

From the speed shot test data, both hydraulic and pneumatic ram designs show potential to have a very accurate pressure velocity relationship. To separate the two systems, different aspects of the ram were considered to determine which ram had the advantage for that respective category:

Table 3: Accelerator Comparison

Accelerator Aspect	Advantage	Reason
Maintenance/ Upkeep	**Pneumatic**	Less complex because maintenance can be performed without removing and storing a compressible fluid.
Breakdown Precautions	**Pneumatic**	With compressible fluids, it is necessary to have precautions in place in the event of a leak in the system.
Footprint	**Pneumatic**	Smaller footprint because there is no need for a pump, a large accumulator, or a fluid reservoir.
Sound Intensity	**Hydraulic**	Compressible fluids are quieter than open-field air release systems.
Building Strain	**Hydraulic**	If a controller could be used, motion profiles can reduce jerk during the acceleration phase.
Safety	**Hydraulic**	Compressible fluids do not have the same potential for explosion at high pressures as compressed air.
Cost	**Pneumatic**	The complexity of a hydraulic system results in higher expenses than a simple air release system.
Longevity	**Pneumatic**	It is yet to be determined which system will have a higher longevity. However, pneumatic rams require less maintenance.
Compatibility/ Versatility	**Hydraulic**	Offers a wider range of operating forces with a potential for a constant force. (However, pneumatic rams do not require as high pressure lines).
Accessibility	**Pneumatic**	The IBRL lab limits the size and complexity of designs because the lab is on the second floor.

Both types of actuators would be sufficient for the purposes of the IBRL. Regardless of the system chosen, a well-designed impactor should offer the accuracy and flexibility necessary for the IBRL. However, because of the added cost and complexity of a custom hydraulic system similar to the impactor at VRTC, it was decided that the benefits did not outweigh the cost.

Cavitating Venturi

A cavitating venturi meter was the most novel accelerator option considered for the ram. These flow meters are used in situations where a constant flow is required. In these applications, the most important characteristic of the flow is the flow rate, and the force output will adjust to counteract the variable resistance force. This pressure force will increase up to 85% of the charge pressure-vapor pressure, at which point the venture meter will stall and no longer provide additional force. To determine the applicability of this system, a background search was conducted on the advantages and disadvantages of the system. The results are outlined below.

Advantages

The most significant advantage for the cavitating venturi lies in the constant velocity characteristics of the meter. Other advantages include a simple design with few moving parts, an available crushing force of up to 85% of the inlet pressure-vapor pressure, and potentially excellent accuracy through a wide range of speeds because of the fact that large pressure variations in the tank correspond to small final velocity variations. If this design were to be utilized, a throat to cylinder diameter ratio would be chosen such that the maximum available pressure would be utilized to achieve the 10 m/s maximum speed (Figure 8). This would allow for improved accuracy at lower velocities.

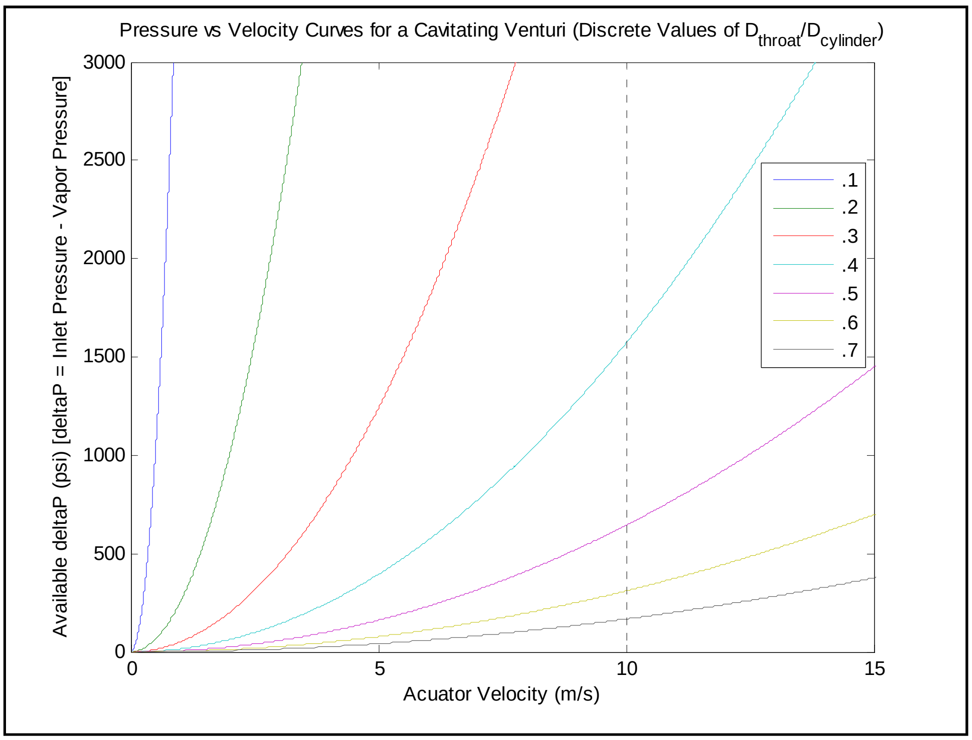


Figure 8: Velocity Curves for Varying Accumulator Pressure Values

Disadvantages

There were several disadvantages to this design that were considered before too many resources were invested into this system. The largest deterrent to this design has been the variable force provided by the accumulator that essentially guarantees the cylinder will reach a pre-determined final velocity. This means that the meter will not be able to provide a constant force over a short time without removing the venturi flow meter and replacing it with a constant area tube. Another potential solution to this problem would be to "stall" the meter by providing an accumulator pressure that was small enough to prevent cavitation. However, there is limited data about this method of stalling the meter and no guarantees that it will work as anticipated. Another issue with this setup relates to the water that the cylinder would use as a medium which would require that the lab not only take preventative measures for leak prevention, but also that the entire circuit be maintained so that it would be free of rust or buildup. Even stainless steel accumulates rust deposits, so maintenance would be more time intensive. Safety issues are also more of a concern even over a pneumatic cylinder because the accelerating cylinder pressurizes as the tank pressurizes. This would mean that a fail-safe mechanical release mechanism would need to be utilized. Even more concerning is the high pressures that are required to reach the desired final velocities while still maintaining accuracy at lower speeds. Although these pressures are in the same range as the hydraulic cylinder discussed previously, the energy storage mechanism in this case would be nitrogen, much more dangerous than compressible fluid. As a final drawback, the accumulator tank would be very large and would be required to be coaxial with the cylinder, which would limit the prospect of incorporating variable height or firing angle into the design.

Conclusion

After a thorough analysis, it was decided that there were too many disadvantages to the system to further consider implementation. The rigidity of the system coupled with the safety issues are enough of a deterrent for further analysis at this point.

Linear Induction Motor

Linear motors were considered for their excellent feedback characteristics and accuracy in obtaining motion profiles. There were several motor types considered including, linear induction motors (iron core and iron-less), tubular linear induction motors, and linear synchronous motors. After a thorough vendor search, one company stood out because they had the highest force linear motors available and offered the motors in a complete package. Anorad Corporation has been a leader in the linear motion industry and appeared to be the best chance of utilizing a linear motor for the impact ram.

Advantages

Linear induction motors (LIM) offer the most controlled motion profiles of all of the accelerators up to this point. With the aid of controllers and positioning systems, LIM's could smooth the acceleration of the system and decrease the jerk to a safer value. Pneumatic cylinders with accumulators act similarly to cannons with very high initial forces (acceleration) that drop off significantly as the cylinder begins to change position (velocity increases, pressure decreases). This puts very high strains on the building that would be reduced with linear motors. (See Rockwell Automation document in Appendix B). Additionally, linear induction motors have the potential to be lower maintenance, and easier to control because of the controller that would come with the system. These systems have a very low noise output, and maintenance would be reduced because the working medium would be readily available electricity instead of gas or liquid.

Disadvantages

The single greatest disadvantage to this system would have been the risk associated with implementing an accelerator that had never been used in a similar situation. After speaking with industry representatives, it became apparent that this was a relatively new and novel technology that still was experiencing growing pains. Because of this, it was initially determined that a commercial off-the-shelf system would be the best route for the IBRL although it was later found that motor performance would be a concern. Additionally, the setup may have been difficult because of the size and shape of the motor. A release mechanism would have been required to prevent the motor from impacting the motor coil on its return path after impact. Figure 9 illustrates the relative design of the motor.

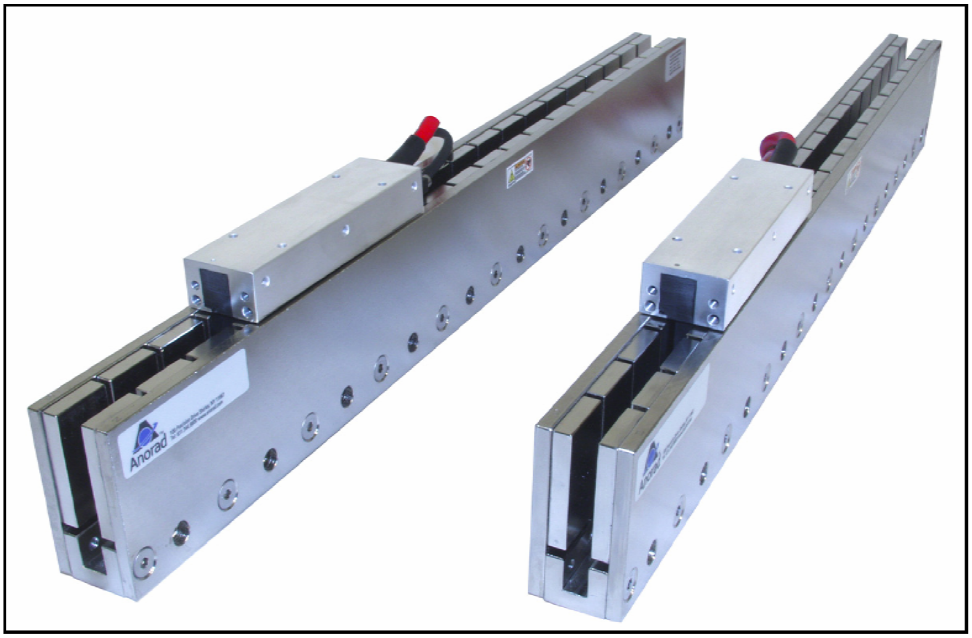

[3]

Figure 9: Anorad Iron-Less Linear Induction Motor

Motor Ability

After a thorough discussion with engineers from Anorad, it was determined that their iron-less LIM would not be sufficient to meet the accelerator requirements. Though their capabilities came very close, they did not feel confident that their motors could reliably provide the required forces, especially if a motion profile were to be implemented with a variable acceleration. A document from Anorad that details the motion profiles and motor capabilities can be found in Appendix B. The document compares the motor capabilities of their largest iron-less LIM to the requirements for the impact ram. Though the motor appears to have been able to meet the requirements, the Anorad representative did not feel comfortable operating so close to the working envelop of the motors.

Conclusion

Though there may be potential for LIM's in the future, too much research and expense would be involved to effectively build a custom LIM system capable of meeting the requirements of the IBRL.

Motion Controlled Hydraulic Circuit

After the review of the charge/discharge hydraulic accelerator at VRTC, a hydraulic cylinder with a controller and proportional valve was considered because of their comparable performance relative to linear motors. Hydraulic motion control circuits offer many of the advantages as linear motor systems and have been widely developed and researched in the past. It was known that the requirements of the accelerator for the IBRL would push the performance envelop of most hydraulic systems in terms of speed and acceleration, but with a modest budget, there was potential to purchase a system at a reasonable cost.

Advantages

The advantages of a controlled hydraulic circuit include all of those mentioned in the above section with the performance analysis of the accelerator from VRTC, with the added luxury of a motion profile. Additionally, if purchased from an outside supplier, there would be less system troubleshooting with the circuit because of the industry experience offered by the vendor. If a custom system were to be built similar to the system at VRTC, it would be more difficult to find an industry representative to service a product that they did not sell. Along those same lines, parts and fittings would not need to be custom manufactured because more commercial off the shelf parts would be available through the service provider.

Disadvantages

As with the advantages, the disadvantages of a motion controlled also closely mirror those of the hydraulic ram from VRTC. Because of the increased capability of this system, complexity also increases which would mean that valuable lab space would be consumed for a pump, an accumulator, and a controller. One disadvantage that would be inherent with a motion controlled system, but not with a system similar to the hydraulic ram at VRTC would be the cost that will be elevated with the high level of sophistication of motion controlled systems because the capabilities of typical hydraulic systems are not designed for such high forces and velocities. The high flow rates corresponding to these velocities introduce a large amount of mechanical shock into the system which requires heavy duty cylinders and multiple proportional valves for fluid control which increases the price of the hydraulic systems significantly. Other potential areas of concern include the mounting of the cylinder on a mobile horizontal beam, and the potential requirement to move the hydraulic system for different testing situations. This may require flexible hydraulic lines, or at least lines that are located to account for the movement of the cylinder.

Commercial Off-the-Shelf

A local supplier was initially contacted to purchase a motion controlled hydraulic accelerator. After an analysis done by hydraulics specialists from Parker Hannifin, the company decided that the shock verification was too high for the capabilities of the Parker cylinders. Parker has long been a leader in hydraulic cylinders, so it was determined that a custom engineering company would be more likely to provide the required system.

Custom System

MTS Systems has been in the testing solutions and motion control industries since 1966, and has provided testing solutions for VRTC in the past. A sale representative from MTS confirmed that they would in fact be able to provide a solution, but that the cost would be significantly higher than originally estimated. The system quoted by MTS appeared to surpass the requirements of the lab with excellent closed-loop feedback control, and a hydraulic pump as quite as a copier machine. However, the large system size and high price of the system prevented implementation. The price for the MTS system exceeded $200,000 [5].

Final Decision

After considering several different accelerator options, no system could be found that could offer a significant improvement over the current system without incurring excessively high costs. Conversations with hydraulics industry representatives have supported the notion that any custom hydraulic system will cost a minimum of $100,000, and will approach $200,000 for a turnkey system with smoothing and motion profiles. Based on these findings, no solution was found for the accelerator portion of the hydraulic ram. Any increase in repeatability for the new ram will have to be realized through the bearing portion of the ram.

Other Considerations

It may be possible that a standard servomotor (1 or 2) in conjunction with an idler sprocket or pulley would be able to provide the linear motion required to accelerate the motor to the desired speeds. The current test frame would be plenty strong to handle a motion profile, and could be utilized with this solution. Due to time constraints, this option was not pursued.

Final Design

Accelerator

As was stated earlier, this component of the impact ram will remain the same. The only significant change in this portion of the ram will be the mounting conditions of the pneumatic cylinder which will be lower than the current ram to minimize the bending moment exerted on the test frame by the accelerator. Mounting the cylinder closer to the central axis will not exert as much force on the linear guides. Additionally, changes may need to be made to the accumulator so that it will not interfere with other parts when the ram is lowered to the lowest position of 18”. The changes will all be related to rotating the cylinder 90° counterclockwise and will include the rotation of the cylinder and a potential mounting bracket for support. The position change is demonstrated in Figure 10 below.

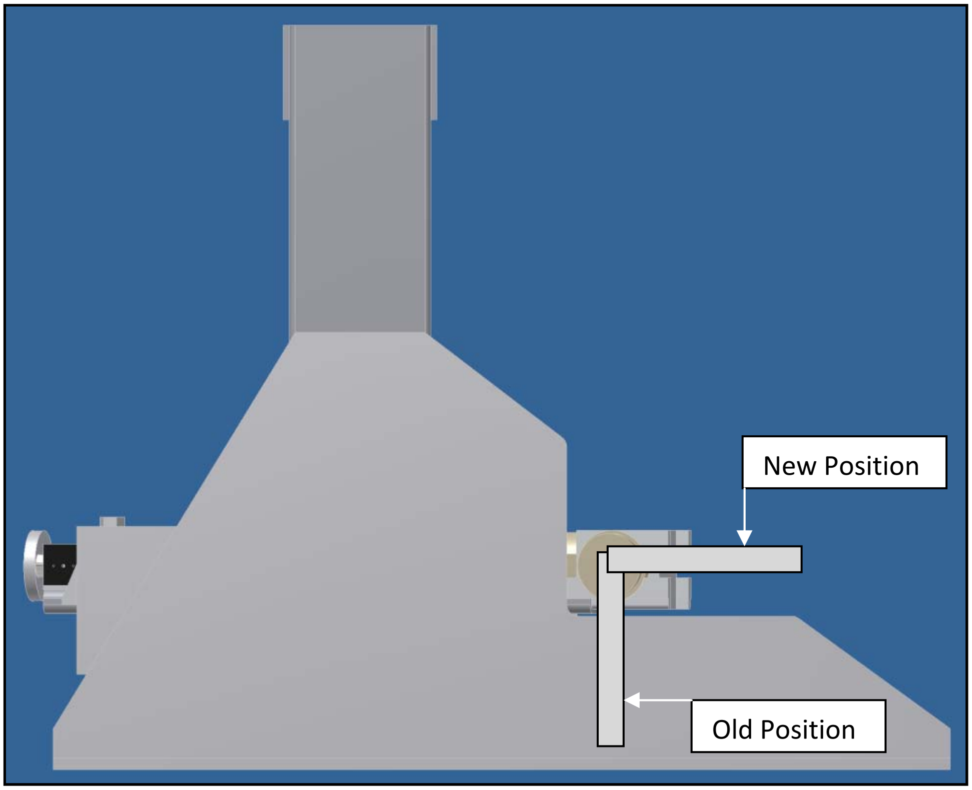


Figure 10: Side View of Ram to Demonstrate Accumulator Relative Position

Bearings

For this segment of the impactor design, there were essentially three different solutions to realistically reduce the friction without creating an excessive amount of noise in the accelerometer data. The first and simplest design would feature plain bearings similar to what has been used up to this point. However, plain bearings incorporate unpredictable static friction, and would not be a significant improvement on the current system. The second solution considered was a guide that would utilize ball bearings. If this design were to be used, a solution would have to have been incorporated to dampen the high frequency noise emitted by the bearings. Because this would have been a difficult task, air bearings were considered with their essentially frictionless and noiseless operation. With recent advances in air bearing technology, air bearings are now manufactured with improved stiffness and accuracy capabilities.

Air Bearings

The final design of the impact ram features linear bearings from New Way Air Bearings. 2" bearings were chosen because they offered the highest stiffness and radial force capabilities while still falling within the weight requirements for the shaft of the impact ram assembly. A moment analysis was conducted on the impact ram to determine the maximum radial force that could be placed on the impact ram without causing the bearings to contact the shafts. Based on this analysis, when the ram is fully extended and loaded with the maximum mass of 37.5 kg (Figure 11), 85.6 lbs of force could be applied to the front end of the shaft before it would contact. This and other values can be seen in Figure 12.

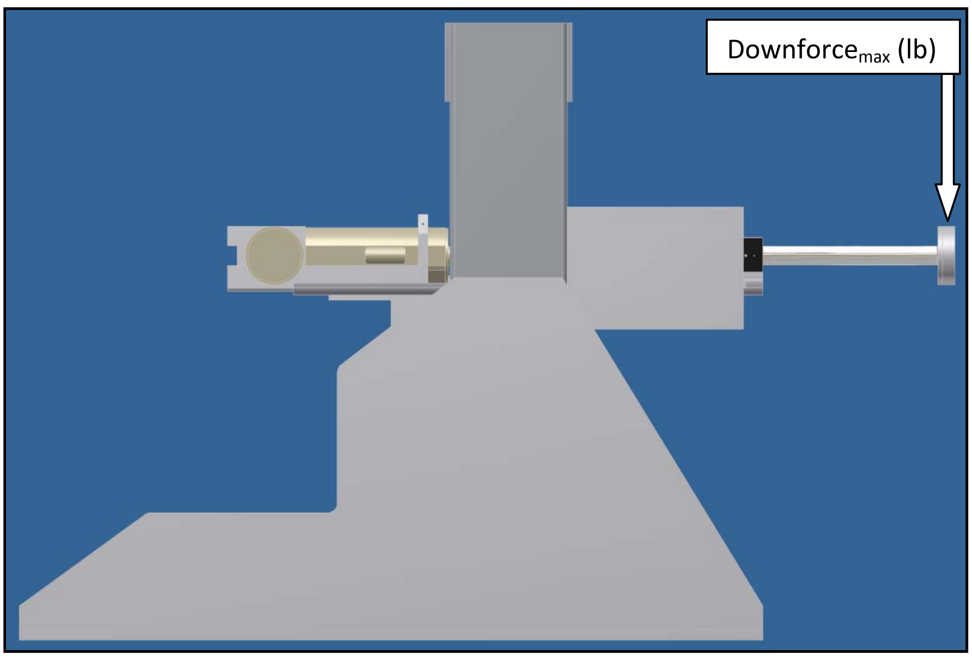


Figure 11: Location of Down force Applied to Ram (Worst Case Scenario)

These values were calculated assuming that the supply pressure for the bearings was 80 psi when, in reality, the bearings can safely handle 100 psi. However, literature data only offers radial loads up to 80 psi so no attempt was made to extrapolate values beyond the conservative operating range because of the uncertainty in radial maximum loads after 80 psi. Additionally, the distance from one end of the bearing to the other was chosen as 10" (8" previously) to allow for greater radial loads. The load analysis script file can be found in Appendix I and Figure 12, Figure 13, and Figure 14 below demonstrate the capabilities of the air bearings. Figure 13 offers the most descriptive visual because these values represent what the operating range of the ram would be for various radial loading conditions. These calculations were performed with the assumption that the radial force would be directed downwards because this was the limiting case.

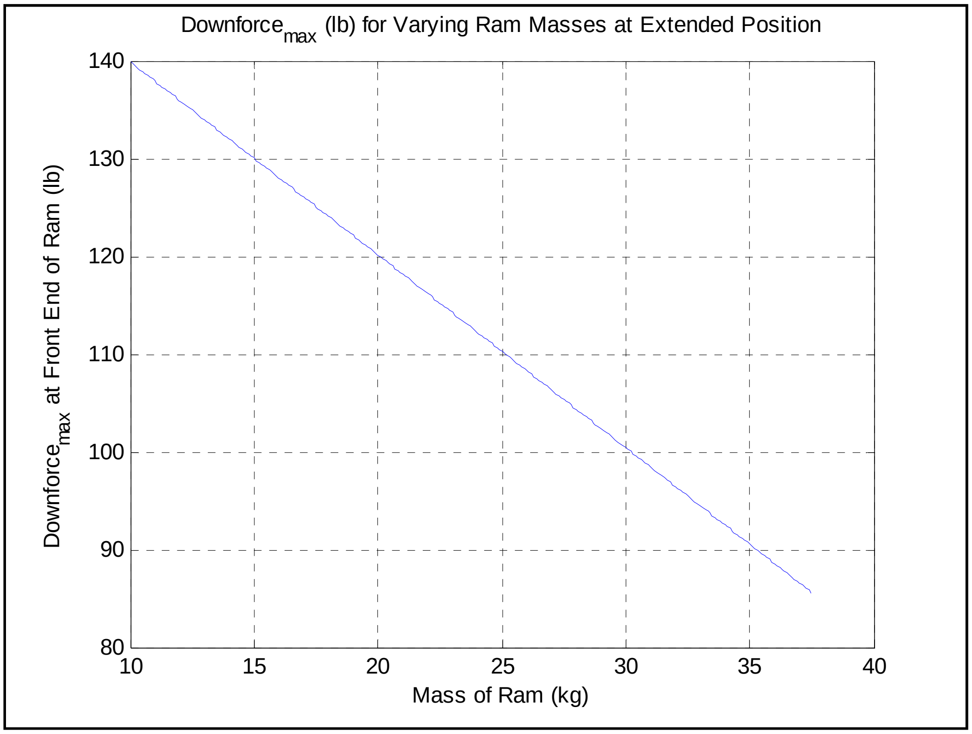


Figure 12: Maximum Down Force at Front of Ram that the Bearings Can Sustain (For Varying Ram Masses)

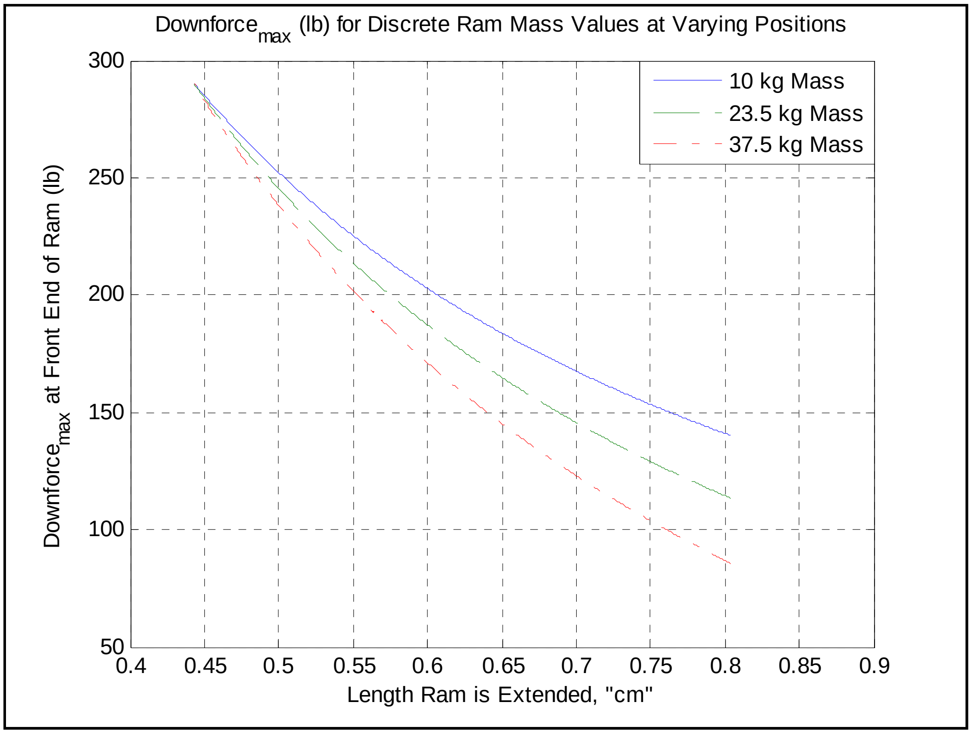


Figure 13: Maximum Down Force at Front of Ram that the Bearings Can Sustain Through Range of Conditions

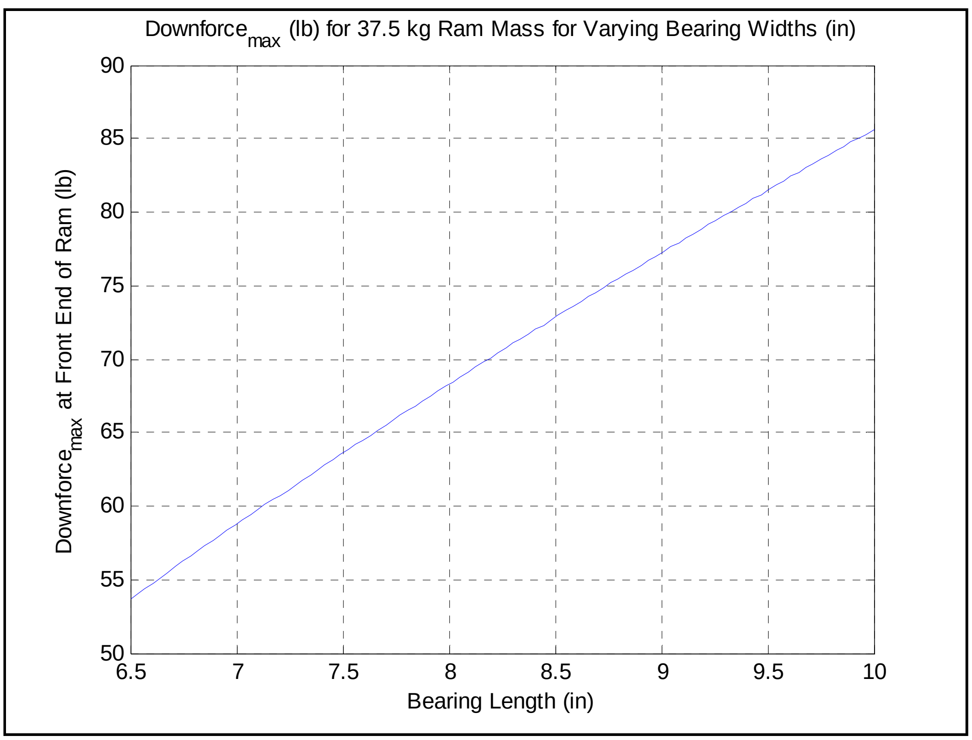


Figure 14: Maximum Down Force at Front of Ram that the Bearings Can Sustain (Varying Bearing Lengths)

It is also notable that significant radial force values above those given in the figures can safely be sustained without damaging the porous media. However, after the maximum value is breached, the contact of the bearing with the shaft will result in discontinuities in friction, and could result in inaccurate data collection. Concern about damage to the bearing is limited because the porous carbon media use in the bearing is resilient and will not fracture with modest loads.

Ram Shafts

Because the impact ram for this project was required to meet a minimum weight of 10 kg, ram shafts were chosen to fall within this weight requirement. Steel was quite a bit heavier than what the design specifications called for, so steel tubing was investigated. However, even hollow steel tubing was heavier than specified, so hard anodized aluminum was chosen because of the low density of these shafts relative to steel. Two 2" shafts will weight approximately 8.39 kg which will allow for about 1.4 kg for the front and rear faces. These shafts were also required to meet class "N" tolerances to work properly when used with air bearings. A detailed drawing of the shaft specifications can be found in Appendix D.

Safety Release Mechanism

Because air bearings will be used on the new impact ram, a safety release mechanism will be required that will prevent the ram from shifting position when the bearings are pressurized. There are several potential solutions to this problem that would be equally effective. Two of those solutions will be discussed here.

The first solution would utilize a permanent magnet to hold the rear face of the ram in the start position. The magnet would be bonded to the front of the accelerator and would have a thickness equal to the length that the plunger protrudes from the pneumatic cylinder (~5/16"). This would be the most cost effective solution as long as the permanent magnet field would not interfere with any instrumentation.

The other solution would incorporate a ball spring locating pin located below that the impact ram. This would allow the ram to be pushed back and latch into the initial position before firing. The only downside to this design would result if the geometry of the rear face plate would alter because of the different sizes that would be required to achieve the desired weight.

Static Support Structure

To develop a resilient and stiff support structure, similar impactors were researched that incorporated vertical adjustment so that a proven design could be utilized for the test frame in the IBRL. A test frame built by MGA featured a similar design with various aspects that could be utilized in the IBRL lab. However, there are some design limitations and constraints that would require a more efficient design than the impactor built by MGA. Two of the most apparent constraints include size and cost. The MGA ram features a four pillar design that would be heavier than the floor in the IBRL could safely handle. Additionally, this design would require more tubular steel, and would limit the rotational angle of the ram if it were to be used. Because of these constraints, a test frame that features only two pillars was designed.

Initially, considerations were made to treat this portion of the project as a weldment. However, because weldment assemblies are susceptible to deformation during the welding process and often require stress relieving, it was decided that assembling with bolts when possible would be the best option for this portion of the assembly. Assuming that a sufficient number of appropriately sized bolts would be used and that an appropriate preload would be applied, the stiffness should be comparable to a weldment design.

Base Plate

Because the new impact ram will have a higher firing axis than the current impact ram (48” vs. 40”), it will be important to keep the center of gravity as low to the floor as possible. Utilizing a massive base plate will be the first step in lowering the center of gravity of the impact ram test frame. The base plate will weight approximately 820 lbs (1”x76”x38”), and will be approximately the same size of the current setup except that the new base will be approximately 10” wider to support the ram when the firing angle is adjusted. Figure 15 demonstrates the locations for the 2½”x2½”x¼” angle iron that will be used to attached the gussets and the vertical columns. There will also be a section of angle iron in the rear to provide a surface for the seismic buffers to thrust against.

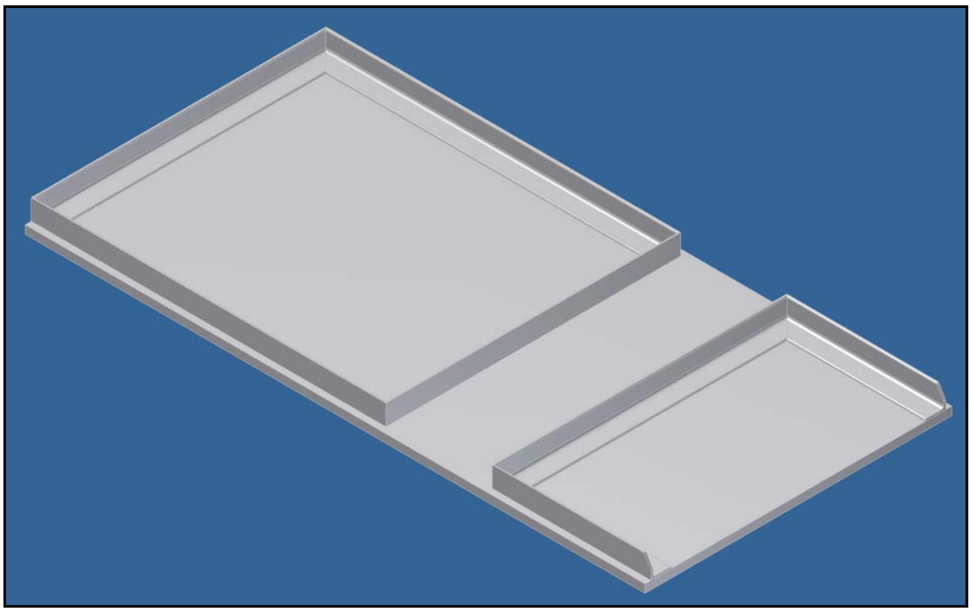


Figure 15: Base Plate with Angle Iron Sections for Bolted Connections

Gussets

The gussets will be required to support the ram primarily along the firing axis. Ideally, the vertical test frame would have been located in the center of the base plate allowing the gussets to have 45° angles, but because of limitations with the carriage assembly, the vertical test frame was required to be located near the front of the ram. A ¾” thickness was chosen for the gussets because this would add additional weight with a low center of gravity. Each gusset weighs approx. 350 lbs for a total of 700 lbs. The gussets will be attached to the base plate and to the vertical support structure via bolted connections using 2½”x2½”x¼” angle iron pieces. The gussets as well as the angle iron supports can be seen in Figure 16.

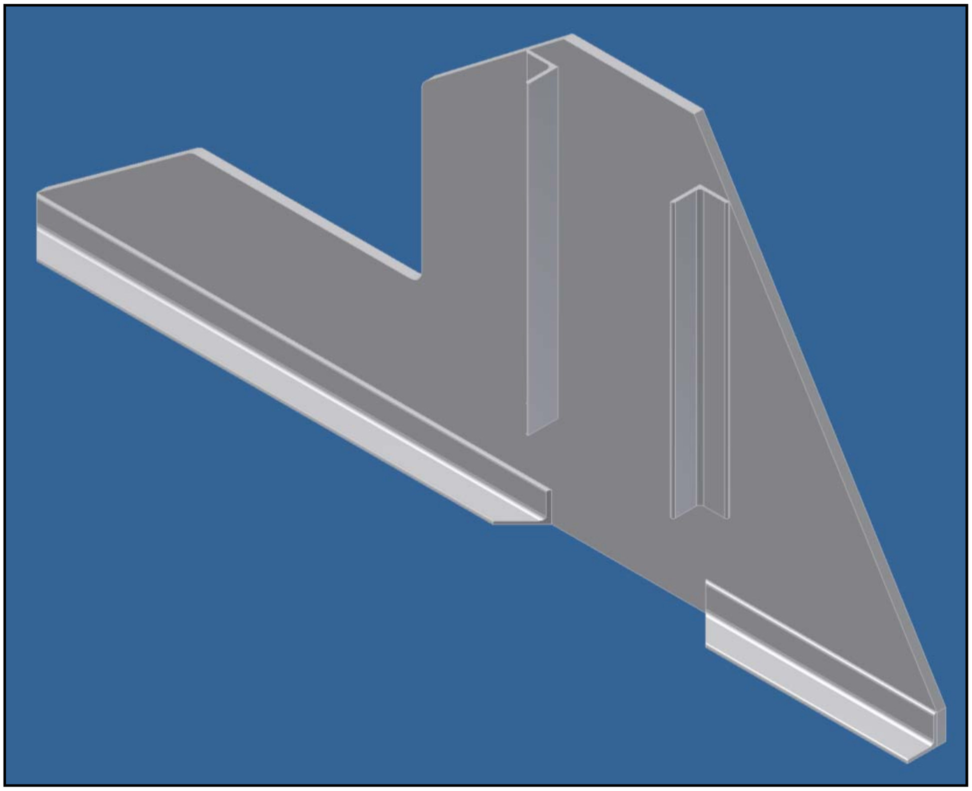


Figure 16: Side Gusset with Angle Iron Sections for Bolted Connections

Linear Guide Assembly (Vertical Motion)

The linear guide assembly will be purchased from a company that manufactures linear motion components because of the experience and tooling required for the mounting of the linear rails and guide blocks. Because the company will be required to invest a significant amount of engineering time into the project, they require a purchase order before developing detailed three dimensional drawings of the product. However, a general drawing of the component (Figure 17) has been developed that provides sufficient detail about the orientation and size of the vertical motion component of the test frame.

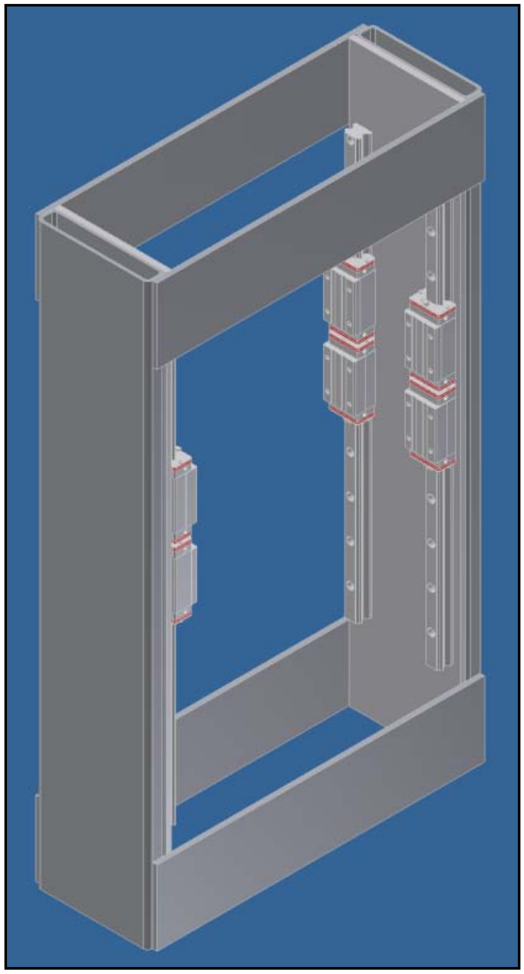


Figure 17: Test Frame (Vertical Component) – Drive Train Not Included

Linear Guideways

The most crucial members of the vertical motion assembly are the linear guideways. Because of the high forces anticipated during impact, the linear guideways will need to have ample stiffness to eliminate jarring in the system. Hiwin is a manufacturer of heavy duty linear guideways that utilize ball bearings with hardened steel tracks to maintain very high stiffness. The design will also include 8 linear guide blocks (4 per side) so that each individual block will not have to bear a moment. A four block pattern as seen below will convert the overall moment force into direct linear forces.

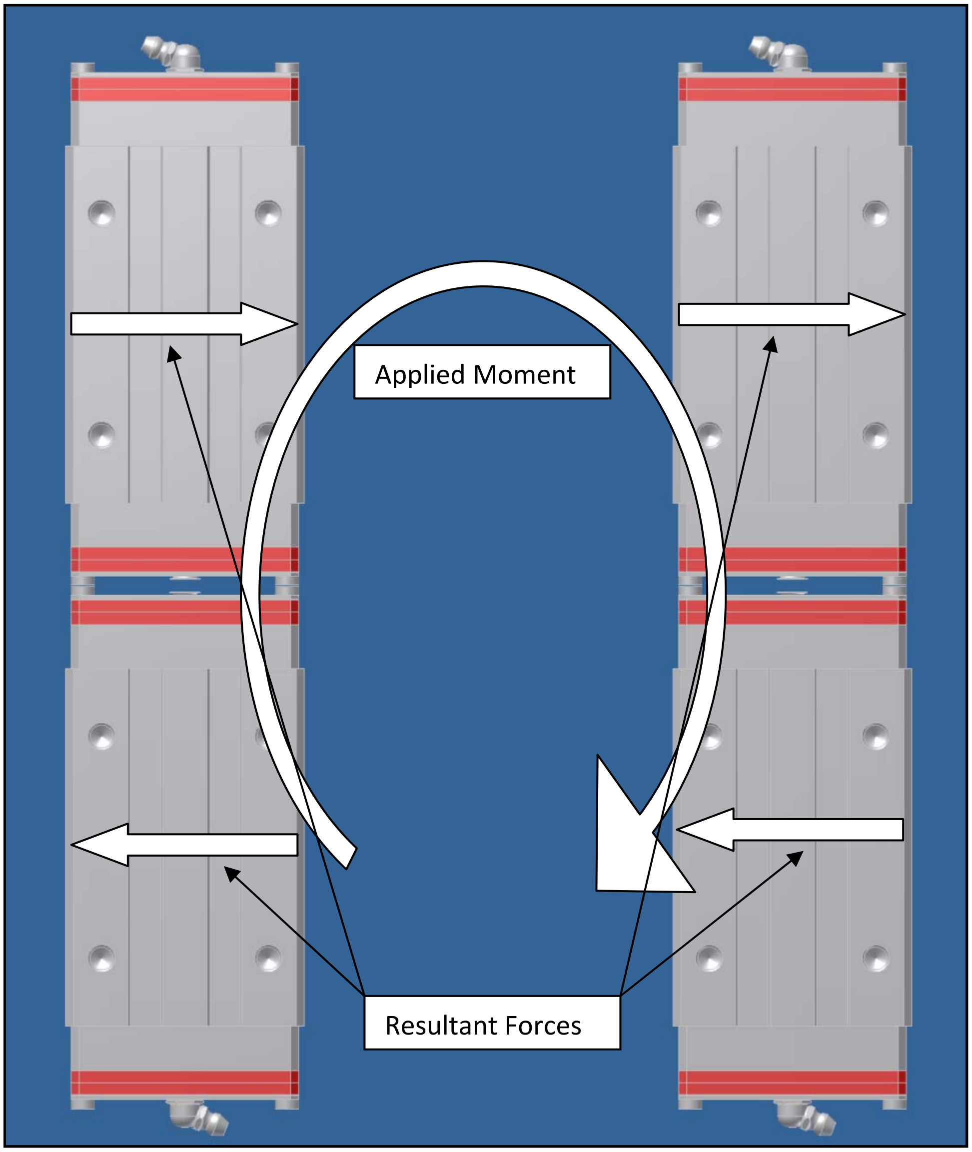


Figure 18: Applied Moment Force and Resultant Linear Forces

The guideways chosen were the LGH 45HA which have a dynamic load rating of 82,656 N (Appendix F). When 8 linear guide blocks are used together, this value rises to 661,249 N. Though this value may seem higher than necessary according to Figure 6 in the Design Specifications and Constraints section, the heavy impact loading experienced during testing cannot be accurately estimated. Providing oversized rails will lower the chances of damage to the rails or guide blocks.

Vertical Test Frame Drive Train

Figure 17 deviates from the actual linear guide assembly because it does not detail the vertical motion unit. This is primarily because the vertical motion unit has yet to be determined. The vendor initially recommended utilizing a threaded rod with a hand crank or motor to obtain the motion, but after further consideration, a hand winch may be a better option because of the cost savings that could be realized. When MotionUSA quoted the project, they included four hand brakes capable of 450 lbs of holding force. By utilizing hand brakes, it would be possible to adjust the location of the carriage assembly by using a hand winch which would be more cost effective than a threaded rod assembly. Figure 19 demonstrates a worm gear winch that could be mounted at the top of the ram potentially in parallel with an identical device to lift the ram along the linear guides. Although this would not provide excellent vertical support, the hand brakes should compensate for the lack of stiffness provided by the cable from the hand winch.

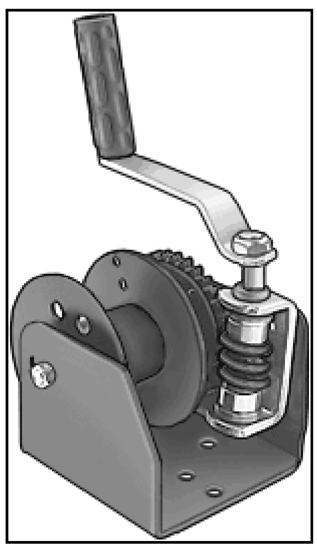

[4]

Figure 19: Hand Winch

Carriage Assembly (Rotational Motion)

Because of the large amount of engineering calculations that will need to be made on the vendor's behalf, exact specifications for the vertical motion linear guide assembly cannot be determined until a purchase order has been issued. Nevertheless, the carriage assembly will fit within a predetermined working envelope, and most dimensional specifications can be confirmed at this point. Still, during the building of the support structure, it is suggested that the construction of the carriage assembly box not be completed until verified product drawings are received from the vendor for the custom linear guide assembly.

The carriage assembly can be seen in Figure 20 and was designed to offer a combination of high structural rigidity and excellent accessibility for alterations on the instrumentation or ram assembly. 4”x3”x3/8” angle iron members were chosen to connect the cross members to the side plates because they would provide a rigid connection between the plates. The angle iron will be welded to the side plates and to the bottom cross members, but the top cross member will use a bolted connection in the event that bearings would need to be removed or adjusted. This would also allow for a potential preload to be applied to the rotational member to further reduce shock in the system.

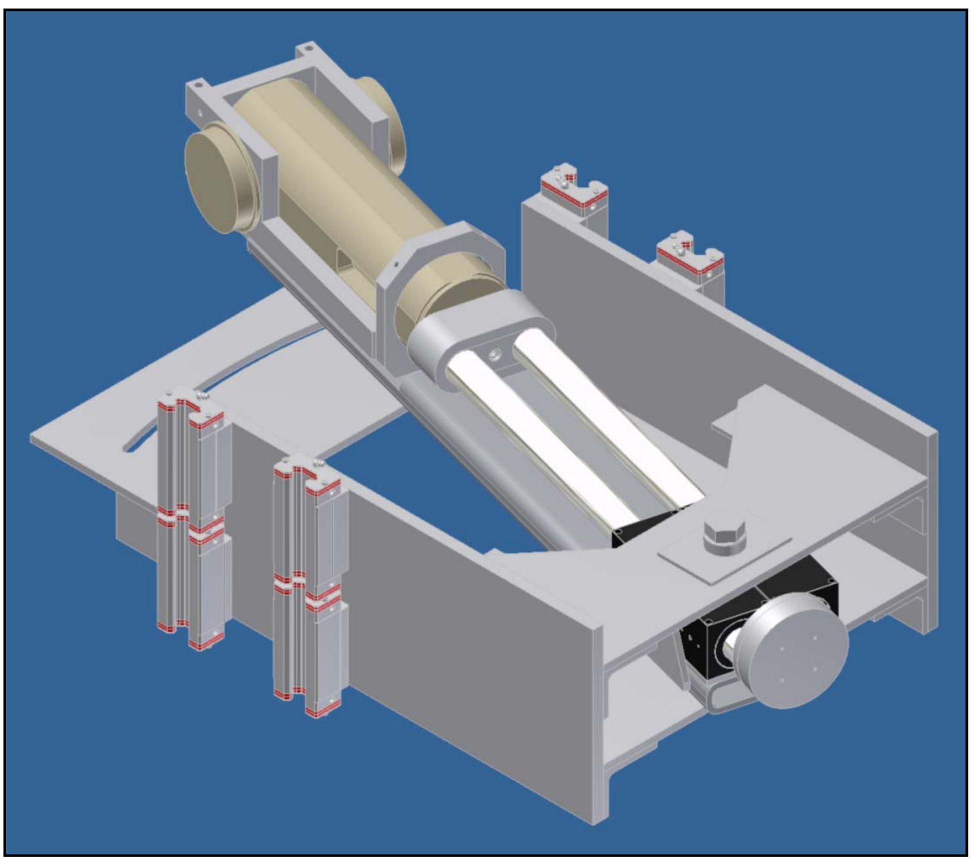


Figure 20: Carriage Assembly (Shown With Guide Blocks)

The front-top crossbeam will have material removed from the backside for visibility of the ram assembly. This will not be required on the bottom crossbeam because visibility from the bottom of the ram is not required. No crossbeam was included on the rear-top of the ram because visibility and clearance are especially important near the accelerator. The use of angle iron will add sufficient strength to allow for this setup.

The connection between the linear guide blocks and the side plates will be joined with 10 mm machine screws. Considering that for each side, 16 machine screws will be used, this connection should not be the weakest link of the ram design. Location of the machine screws has been a concern, but according to the final drawings, sufficient clearance should be available for the installation of these screws. The bottom row of machine screws will have to be installed through the angle iron and the side plates, but this is not anticipated to be a problem.

Pin Joint

The pin joint of the carriage assembly is estimated to be the weakest link in the engineering design with respect to noise within the system. Because all of the force of the ram will be transmitted through this joint, considerable care should be taken in maintaining as tight of a tolerance as possible. This design will incorporate a threaded pin so that the assembly can be clamped down to further isolate the joint. A simple shear calculation was performed in Appendix H to verify that the pin does have a large enough diameter. Additionally, Nominal Failure Load in Double Shear for a typical 1" diameter clevis pin is 806,361 N which is significantly higher than the expected forces from the accelerator [6]. In fact, this would introduce a factor of safety of approximately 22.

Figure 21 below shows the orientation of the parts related to the rotational motion within the carriage assembly. The clevis pin support tube connects the bearing top plate, bearing bottom plate, and the horizontal rectangular tube steel into one piece that will put minimal strain on the clevis pin. All of these pieces will be welded for maximum strength because using mechanical fasteners for such small pieces would not be as solid. In fact, utilizing a clevis pin support tube will turn the clevis pin into a simple double shear problem.

To further strengthen the top and bottom plates, clevis pin end mount support blocks were added. This will increase the effective thickness from ½” to 1 ½” for the upper and lower plates. This was primarily done to prevent wear because with heavy impact loads, even a small amount of vibration in the system will lead to inaccuracies in data collection.

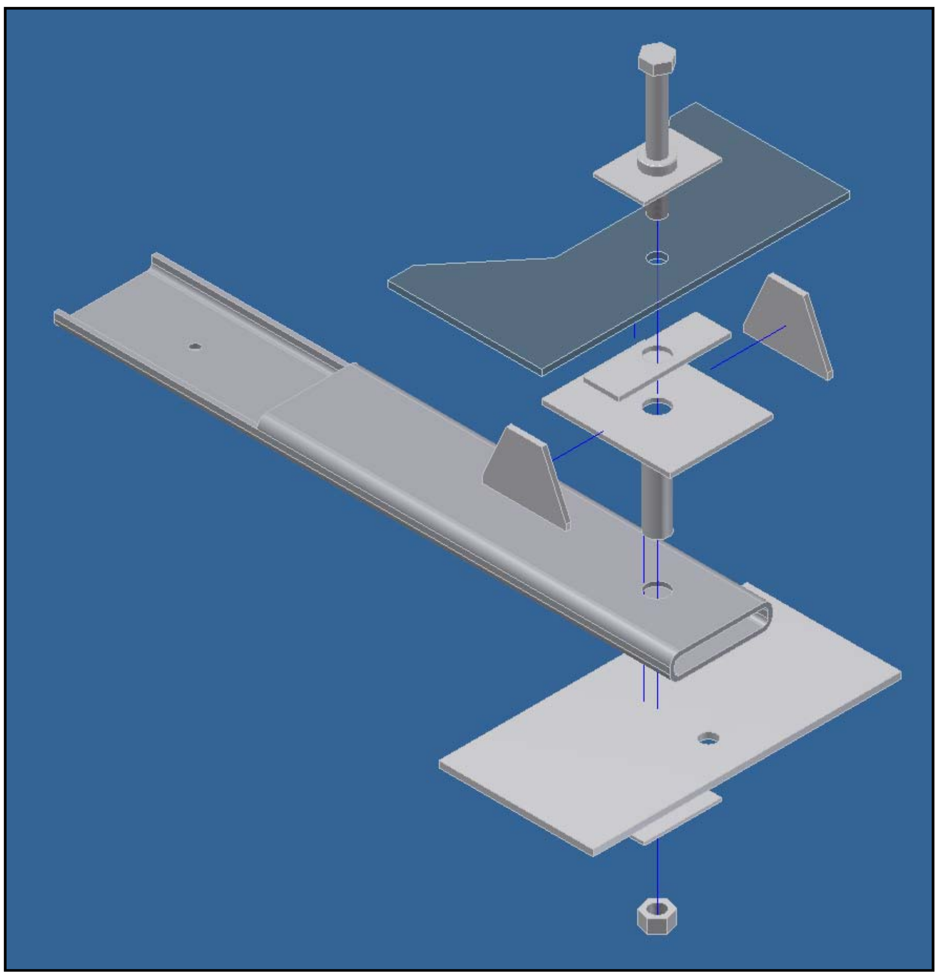


Figure 21: Carriage Assembly-Exploded View (Some Items Not Shown)

Ram Faces

The current ram utilizes four different ram faces during typical impacts that attempt to match a weight of 24 kg. This weight requirement will be met with the new ram by adding weights to the front and the back of the ram via steel plates. If the current initial ram weights 10 kg, and required components that weight approximately 2.84 kg (aluminum impact face, load cell, accelerometers, screws), a total of 11.8 kg would be necessary via ram weights to achieve the total weight of 24 kg. These faces are included in the budget. As an aside, to minimize noise and vibration during testing, it is recommended that a thin gasket be placed between the ram faces and weights.

Budget

The services required portion of the budget was estimated with limited experience in machine work and cost estimating. Some of the costs could be much higher if the parts need to be sent to a third party machine shop for work, and some may be lower if they are able to be performed in house at cost. Before the project is implemented, it is recommended that the cost analysis be verified so that it can be known where the work will be performed, and at what cost it will be done. Below is a list of considerations that may affect the pricing of some parts:

- Gussets were quoted before material was removed to provide room for accumulator. It is likely that this will add a small cost to their manufacturing.
- Mechanical fasteners were not estimated in most cases unless they were estimated to be costly.
- There could be a significant savings (~$5,000) in the vertical test frame if a hand winch were used instead of the quoted drive train featuring a threaded rod and a motor or gearbox.
- The service portion of the noise control cost analysis was estimated to be the same cost of the materials at the recommendation of a sales representative. (Doubled materials cost).

Table 4: Static Support Structure

		STATIC SUPPORT STRUCTURE	ASSEMBLY NUMBER: 101					
ITEM	QTY	DESCRIPTION	MATERIAL	WEIGHT (LBS)	TOTAL WT (LBS)	COST	TOTAL COST	VENDOR
001	1	1" BASE PLATE (76"x38")	A-36 HRS	819.33	819.33	$899.00	$899.00	ALRO
002	2	GUSSET (PER PRINT)	A-36 HRS	329.45	658.89	$609.75	$1,219.50	ALRO
003	2	BOTTOM-REAR-LONGITUDINAL ANGLE IRON (2.5"X2.5"X1/4")	A-36 HRS	13.93	27.87	-	-	ALRO
004	2	BOTTOM-FRONT LONGITUDINAL ANGLE IRON (2.5"X2.5"X1/4")	A-36 HRS	6.20	12.41	-	-	ALRO
005	3	BOTTOM-LATERAL ANGLE IRON (2-1/2"X2-1/2"X1/4")	A-36 HRS	11.43	34.30	$127.10	$127.10	ALRO
006	25	BOLT (3/4"-16 1-1/4") P/N 92865A472	STEEL	0.11	2.75	$1.57	$39.30	MCMASTER
		MATERIALS GRAND TOTAL			1,555.54		**$2,284.90**	

ITEM	TIME	SERVICES REQUIRED	SRVC CDE	COST	VENDOR
001		DRILL/TAP HOLES IN BASE PLATE FOR ANGLE IRON	201	$80.00	TRC
001		DRILL HOLES IN BASE PLATE FOR ANCHOR BOLTS	202	$40.00	TRC
002		DRILL HOLES FOR ANGLE IRON/VERTICAL TEST FRAME	203	$80.00	TRC
002		REMOVE MATERIAL	204	$100.00	TRC
101		PAINT	205	$100.00	TRC
		SERVICE GRAND TOTAL		**$400.00**	

GRAND TOTAL		**$2,684.90**

Table 5: Test Frame (Vertical)

		TEST FRAME (VERTICAL)	ASSEMBLY NUMBER: 102					
ITEM	QTY	DESCRIPTION	MATERIAL	WEIGHT (LBS)	TOTAL WT (LBS)	COST	TOTAL COST	VENDOR
007	1	VERTICAL TEST FRAME	STEEL	708.75	708.75	$12,500.00	$12,500.00	MOTIONUSA
008	1	VERTICAL MOTION UNIT	VARIOUS	100.00	100.00	$5,000.00	$5,000.00	MOTIONUSA
		MATERIALS GRAND TOTAL			808.75		**$17,500.00**	

ITEM	TIME	SERVICES REQUIRED	SRVC CDE	COST	VENDOR
		SERVICE GRAND TOTAL		**$0.00**	

GRAND TOTAL		**$17,500.00**

Table 6: Carriage Assembly

CARRIAGE ASSEMBLY ASSEMBLY NUMBER: 103

ITEM	QTY	DESCRIPTION	MATERIAL	WEIGHT (LBS)	TOTAL WT (LBS)	COST	TOTAL COST	VENDOR
009	2	3/4" SIDE PLATES (12-1/2"X30")	A-36 STEEL	79.79	159.58	$95.25	$190.50	ALRO
010	2	36" BOTTOM ANGLE IRON (4"X3"X3/8")	A-36 STEEL	25.77	51.54	$31.59	$63.19	ALRO
011	2	12" TOP ANGLE IRON (4"X3"X3/8")	A-36 STEEL	8.59	17.18	$10.53	$21.06	ALRO
012	1	1/2" CROSS BEAM TOP FRONT (12"X24")	A-36 STEEL	30.75	30.75	$46.25	$46.25	ALRO
013	1	1/2" CROSS BEAM BOTTOM FRONT (8"X24")	A-36 STEEL	40.85	40.85	$46.25	$46.25	ALRO
014	1	1/2" CROSS BEAM BOTTOM REAR (12"X24")	A-36 STEEL	38.78	38.78	$46.25	$46.25	ALRO
015	1	48" RECTANGULAR TUBING (8"X2"X3/8")	ASTM A-500	89.48	89.48	$126.75	$126.75	ALRO
016	1	1/2" BEARING MOUNTING BOTTOM PLATE (8"X10")	A-36 STEEL	11.35	11.35	$13.44	$13.44	ALRO
017	2	1/2" BEARING SIDE PLATES (4.75"X6.5")	A-36 STEEL	3.26	6.52	$6.39	$12.77	ALRO
018	1	1/2" BEARING MOUNT TOP PLATE (2.75"X8.75")	A-36 STEEL	3.07	3.07	$4.03	$4.03	ALRO
019	1	1.75"X1.25" CLEVIS PIN SUPPORT TUBE (5.875")	A-36 STEEL	1.96	1.96	$10.00	$10.00	ALRO
020	2	CLEVIS PIN END MOUNT SUPPORT BLOCK	A-36 STEEL	2.02	4.04	$20.00	$40.00	ALRO
021	1	THREADED CLEVIS PIN	CRS	11.14	11.14	$10.00	$10.00	ALRO
022	6	BOLTS FOR CARRIAGE PLATE (TOP-FRONT)	STEEL	5.00	30.00	$2.00	$12.00	MCMASTER
023	1	SOCKET CAP HOLD DOWN SCREW (3/4"-16, 1-1/4")	STEEL	1.17	1.17	$2.39	$2.39	MCMASTER
024	1	ACCELERATOR FRONT MOUNT	A-36 STEEL	6.69	6.69	$10.00	$10.00	ALRO
025	2	ACCELERATOR SIDE MOUNTS	A-36 STEEL	19.24	38.48	$10.00	$20.00	ALRO
026	1	ACCELERATOR REAR MOUNTS	A-36 STEEL	10.69	10.69	$10.00	$10.00	ALRO
		MATERIALS GRAND TOTAL			553.27		**$684.89**	

ITEM	TIME	SERVICES REQUIRED	SRVC CDE	COST	VENDOR
103		FACE OFF ALL PARTS	206	$200.00	TRC
009		WELD TO ANGLE IRON	207	$10.00	TRC
010		DRILL HOLES FOR CROSS BEAMS	208	$10.00	TRC
011		DRILL HOLES FOR CROSS BEAMS	209	$10.00	TRC
012		DRILL HOLES FOR ANGLE IRON	210	$10.00	TRC
012		DRILL HOLE FOR CLEVIS PIN	211	$10.00	TRC
012		DRILL HOLES FOR CLEVIS PIN END MOUNT SUPPORT	212	$10.00	TRC
012		REMOVE MATERIAL (FOR VISIBILITY OF BEARINGS)	213	$20.00	TRC
013		WELD TO ANGLE IRON	214	$10.00	TRC
013		DRILL HOLES FOR CLEVIS PIN	215	$10.00	TRC
013		DRILL HOLES FOR CLEVIS PIN END MOUNT SUPPORT	216	$10.00	TRC
014		MACHINE SLOT FOR CLAMP DOWN BOLT	217	$10.00	TRC
014		WELD TO ANGLE IRON	218	$10.00	TRC
015		DRILL HOLES FOR RAM MOUNTS	219	$10.00	TRC
015		REMOVE MATERIAL FOR RAM MOUNTS	220	$20.00	TRC
015		DRILL HOLE FOR CLAMP DOWN BOLT	221	$10.00	TRC
015		DRILL HOLE FOR CLEVIS PIN SUPPORT TUBE (019)	222	$10.00	TRC
015		WELD CLEVIS PIN SUPPORT TUBE (019)	223	$10.00	TRC
015		WELD BEARING MOUNT BOTTOM PLATE (016)	224	$10.00	TRC
015		WELD BEARING SIDE PLATES (017)	225	$10.00	TRC
016		DRILL/TAP HOLES FOR MOUNTING BLOCKS (030)	226	$10.00	TRC
016		WELD BEARING SIDE PLATES (017)	227	$10.00	TRC
016		DRILL HOLE FOR CLEVIS PIN SUPPORT TUBE (019)	228	$10.00	TRC
017		WELD BEARING MOUNT TOP PLATE (018)	229	$10.00	TRC
018		DRILL HOLE FOR CLEVIS PIN SUPPORT TUBE (019)	230	$10.00	TRC
019		WELD BEARING MOUNT BOTTOM PLATE (016)	231	$10.00	TRC
019		WELD BEARING MOUNT TOP PLATE (018)	232	$10.00	TRC
020		MACHINE SUPPORT BLOCK	233	$30.00	TRC
021		MACHINE THREADED CLEVIS PIN	234	$30.00	TRC
024		REMOVE MATERIAL (HOLE/ANGLES)	235	$30.00	TRC
024		DRILL HOLES (LOCATING PINS/TUBE/SIDE MOUNTS)	236	$10.00	TRC
025		MACHINE TO PRINT	237	$60.00	TRC
026		DRILL HOLES (SIDE MOUNT)	238	$10.00	TRC
103		ASSEMBLE/TROUBLESHOOT	239	$500.00	TRC
		SERVICE GRAND TOTAL		**$1,150.00**	

GRAND TOTAL		**$1,150.00**

Table 7: Ram Assembly

		RAM ASSEMBLY	ASSEMBLY NUMBER: 104					
ITEM	QTY	DESCRIPTION	MATERIAL	WEIGHT (LBS)	TOTAL WT (LBS)	COST	TOTAL COST	VENDOR
027	1	ACCELERATOR	ALUMINUM	34.74	34.74	$0.00	$0.00	TRC
028	2	RAM SHAFT (2"X30", CLASS N)	ALUMINUM	9.24	18.47	$495.00	$990.00	MOTIONUSA
029	4	AIR BUSHINGS (2" ID)	ALUMINUM	1.24	4.96	$420.00	$1,680.00	MOTIONUSA
030	4	MOUNTING BLOCKS	ALUMINUM	2.31	9.23	$230.00	$920.00	MOTIONUSA
031	4	SOCKET CAP SCREW (1/2"-13 1-1/2")	STEEL	0.20	0.80	$1.10	$4.40	MCMASTER
032	1	RAM REAR FACEPLATE	ALUMINUM	0.4998	0.50	$30.00	$30.00	ALRO
033	1	RAM FRONT FACEPLATE	ALUMINUM	2.65	2.65	$30.00	$30.00	ALRO
034	1	RAM REAR WEIGHT	STEEL	16.14	16.14	$40.00	$40.00	ALRO
035	1	RAM FRONT WEIGHT	STEEL	9.86	9.86	$40.00	$40.00	ALRO
036	1	FITTING/AIRLINES/COMPRESSOR	BRASS	TBD	TBD	$200.00	$200.00	MOTIONUSA
		MATERIALS GRAND TOTAL			97.35		**$3,934.40**	

ITEM	TIME	SERVICES REQUIRED	SRVC CDE	COST	VENDOR
104		FACE OFF ALL UNFINISHED PLATES	240		
032		MACHINE REAR FACEPLATE	241	$40.00	TRC
033		MACHINE FRONT FACEPLATE	242	$40.00	TRC
034		MACHINE REAR WEIGHT	243	$40.00	TRC
035		MACHINE FRONT WEIGHT	244	$40.00	TRC
036		CONNECT AIR LINES/COMPRESSOR	245	$50.00	IBRL
104		ASSEMBLE BEARING ASSEMBLY	246	$100.00	TRC
		SERVICE GRAND TOTAL		**$310.00**	

GRAND TOTAL		**$4,244.40**

Table 8: Noise Control

		NOISE CONTROL	ASSEMBLY NUMBER: 105			
ITEM	QTY	DESCRIPTION	MATERIAL	COST	TOTAL COST	VENDOR
037	10	RSP 18"X18" WAFFLE PADS	SYNTHETIC	$71.00	$710.00	KINETICS
038	6	HS-1-2000 SNUBBERS	STEEL	$120.00	$720.00	KINETICS
039	12	KCCAB-50-500 ANCHOR BOLTS	STEEL	$5.00	$60.00	KINETICS
040	4	KCCAB-50-700 ANCHOR BOLTS	STEEL	$9.00	$36.00	KINETICS
041	4	TG-50 ISOLATION GROMMETS	VARIOUS	$2.25	$9.00	KINETICS
		MATERIALS GRAND TOTAL			**$1,535.00**	

ITEM	TIME	SERVICES REQUIRED	SRVC CDE	COST	VENDOR
105		INSTALLATION	247	$1,535.00	SAUER INC.
		SERVICE GRAND TOTAL		**$1,535.00**	

GRAND TOTAL		**$3,070.00**

Table 9: Final Design

FINAL DESIGN			
ITEM	DESCRIPTION	TOTAL COST	TOTAL WEIGHT (LBS)
101	STATIC SUPPORT STRUCTURE	$2,684.90	1555.54
102	TEST FRAME (VERTICAL COMPONENT)	$17,500.00	808.75
103	CARRIAGE ASSEMBLY	$1,834.89	553.27
104	RAM ASSEMBLY	$4,244.40	97.35
105	NOISE CONTROL	$3,070.00	N/A

GRAND TOTAL	**$29,334.18**	3014.90
10% INFLATION	**$32,267.60**	

Discussion

There were a couple of concerns regarding ram design that will need to be addressed as the ram is built and tested. The forces that the test frame will experience are dynamic, so certain aspects of the test frame design will need to be analyzed while the test frame is being constructed and tests. A few of these aspects are discussed below.

Side Supports

The current ram does not experience side loading, so the narrow width of the current ram is a concern. Although the isolation grommets from Kinetic Noise Control (Ketchum & Walton Co.) will be installed to prevent overturning, it would be ideal to minimize the force that these concrete anchors are required to sustain. During the initial implementation of the ram, it is recommended that before the ram is anchored to the floor, low energy tests be conducted to verify that the frame does not have a tendency to overturn when shots are fired at an angle. If this is found to be the case, side supports similar to what is shown in Figure 22 should be attached to alleviate some of the strain experienced by the anchor bolts and the concrete floor.

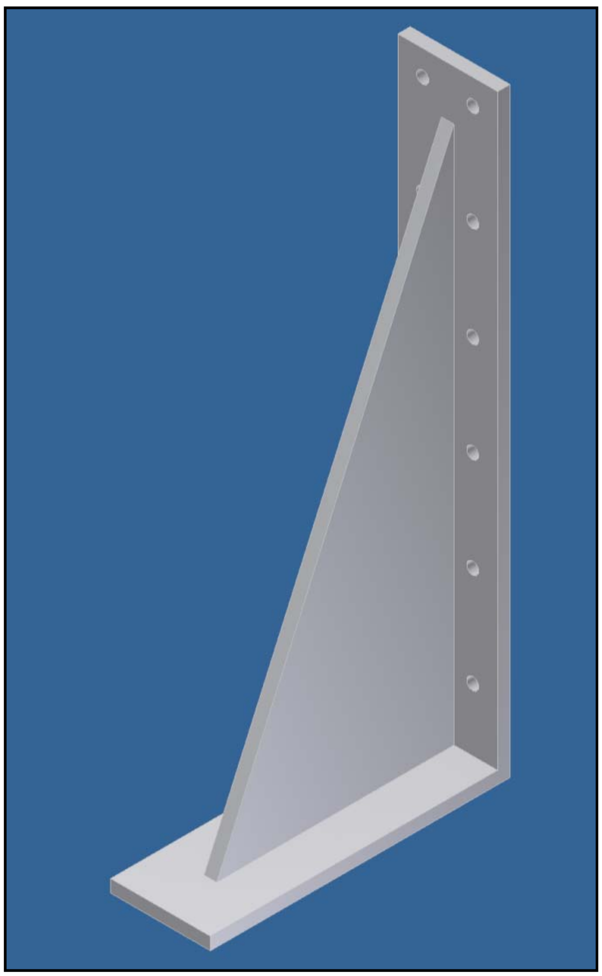


Figure 22: Side Supports

Wings

The other addition that may need to be considered, though not as thoroughly as the side supports would be support wings that would attach to the side of the carriage assembly and would thrust against the vertical tubular steel to prevent jarring in the system when the ram would be fired. These support wings were heavily considered in an earlier design that utilized less durable linear guideways and a narrower section of tubular steel. The intentions were to increase the stiffness of the system and to help counteract the moment created when the ram would be fired. However, because of the heavier built linear rails and the use of 4 guide blocks per side this need has been reduced. A previous design featuring the support wings can be seen in Figure 6.

Weight Plates

Another consideration for the design of the ram lies in the weight distribution of the carriage assembly. The current design appears to be front heavy, so adding weights to the rear of the carriage assembly to balance the weight will need to be considered.

Other Concerns

Areas of the ram where welds are utilized instead of mechanical fasteners will likely need to be stress relieved because weldment can suffer from deformation if not treated properly. Also, the raw steel purchased for this project may need to be machined with an end mill to eliminate variations in surface conditions. As a final consideration, handling of the large portions of the ram would be less difficult of there were threaded holes that eye hooks could attach to. Otherwise, it may be difficult to lift the assemblies, especially those that are heavier such as the base plate and side gussets.

Conclusion

It was the objective of this research project to:

- Determine design requirements of a new ram to achieve impacts at a constant velocity given varying impact ram mass and velocity requirements of current ATD and PMHS tests [1]
- Determine the instrumentation to measure the displacement, velocity, acceleration, and force of the ram, and to consider the possibility of adjusting the height, the angle of impact, and the mass/face of the ram
- Design and document a new impactor to meet the agreed upon design requirements

Although a cost effective capable replacement could not be found for the accelerator portion of the ram, improvements in constant velocity should be realized with the introduction of the air bearings into the system. The current ram has the largest velocity losses at low velocities, so adding air bearings to the system will help the most in this velocity range. Also, because the friction will become negligible, slightly higher final velocities should be reached because the ram will no longer have to overcome the forces of friction during acceleration.

New instrumentation for the ram was largely not determined because the current instrumentation is sufficient in measuring the kinematic characteristics of the ram. Additionally, because the same accelerator will be utilized, most of the current measuring devices can be utilized. Although there was a search performed for a vendor for an inertially compensated load cell, nothing could be found within the time frame that had a high enough of a response for the testing that takes place in the IBRL. However, in the future it may be possible to eliminate inertial affects by utilizing a high pass filter instead of inertial compensation via an estimated mass acceleration calculation. This could only be performed if there was a sizeable difference between the signal frequency of the desired response, and that of the inertia.

The adjustment of the height and firing angle of the ram was designed for by using linear guides and a clevis pin respectively. The test frame was designed sufficiently robust that the safety factor of 2 was not approached. The variable mass of the ram will be realized by creating custom weight plates to be added to the front and the rear as new face plates are developed. Because the current ram almost always uses approximately the same ram weight for testing, multiple weight plates were not developed. However, whenever the need for a different mass arises, the design and construction of weights should be cost effective.

More specifically, the completion of the design specifications is discussed below.

- Achieve a speed of 10 m/s with a mass of 23.5 kg
  - With the reduction in friction, these values will be approached but will probably not be met.
- Achieve an accuracy of ±0.05 m/s
  - As above, the reduction in friction should increase the accuracy. Actual values will not be able to be determined until the system is built and tested.
- Negligible & predictable friction
  - This specification was met with the use of air bearings.
- Low noise
  - This specification was met with the use of air bearings.
- Impact ram axially constrained
  - Because the air bearings will require two shafts, this specification will be met.
- Minimum mass of 10 kg
  - Although a face plate would have to weigh less than .17 kg, this specification was met in technicality.
- Inertially compensated load cell
  - Inertial compensated will continue to occur in the data reduction and not the load cell as previously hoped for.
- Rigid & resilient support structure
  - This specification was met with the oversized linear guides and clevis pin.
- Size and weight of individual components
  - The largest component will be the base plate and will fall under the weight and size requirements.
- Compatible with drop tower/overhead support system
  - Alterations will have to be made so that the drop tower/overhead system will be compatible with the new impact ram.
- Vertical adjustment between 18" and 48"
  - This specification was achieved with the use of the linear guides and vertical motion component. However, for a significant cost savings, more research would need to be performed on the applicability of hand winches for this aspect of the project.
- Angular adjustment ±15°
  - This specification was achieved with the use of the clevis pin and carriage assembly.

Finally, the drawings for the final design of the ram can be found below in Appendix J.

References

1. ISO. (1997) Road Vehicles – anthropomorphic side impact dummy – lateral impact response requirements to assess the biofidelity of the dummy. ISO/DTR 9790-7 Document N455 – Revision 4.

2. Mallory, Linear Impactor Brake Evaluation – Summary, 10/12/2005

3. Image: http://www.rockwellautomation.com/anorad/products/linearmotors/index.html

4. Image: McMaster-Carr Online Catalog, pp. 1420. http://www.mcmaster.com/

5. Sylvester, Rex. MTS Quotation for MTS FlexTest 60 Controller and High Rate Actuator. Sept. 17, 2008.

6. Collins, Mechanical Design of Machine Elements and Machines, John Wiley & Sons, Inc. Nov 6, 2002.

Appendix

A. Speed Shots (IBRL, VRTC)

Tabulated results from ram documentation for the VRTC hydraulic ram, and the IBRL pneumatic ram. Pneumatic Ram Speed Shots (4/17/07).

Table 10: P/V Relationship IBRL Speed Shots (4/17/07)

	psi	Vel_{max}	Vel_{impact}	Vel_{loss}
1	59.6	1.443	0.615	0.828
2	59.6	1.446	0.661	0.785
3	59.6	1.433	0.654	0.779
4	59.6	1.434	0.588	0.846
5	59.6	1.431	0.577	0.854
Std. Dev.	0.000	**0.007**	**0.038**	**0.035**
1	99.7	2.398	2.032	0.366
2	99.7	2.401	2.038	0.363
3	100.2	2.430	2.066	0.364
4	99.7	2.405	2.008	0.397
5	99.7	2.374	1.980	0.394
Std. Dev.	0.224	**0.020**	**0.032**	**0.017**
1	150.0	3.540	3.368	0.172
2	149.9	3.554	3.388	0.166
3	150.0	3.565	3.418	0.147
4	150.0	3.525	3.354	0.171
5	150.0	3.543	3.359	0.184
Std. Dev.	0.045	**0.015**	**0.026**	**0.013**
1	199.8	4.894	4.818	0.076
2	199.9	4.769	4.692	0.077
3	199.8	4.714	4.589	0.125
4	199.8	4.884	4.806	0.078
5	199.8	4.729	4.628	0.101
Std. Dev.	0.045	**0.086**	**0.103**	**0.021**
1	250.2	5.894	5.753	0.141
2	249.8	5.895	5.782	0.113
3	249.7	5.950	5.834	0.116
4	249.7	5.952	5.868	0.084
5	249.7	5.973	5.762	0.211
Std. Dev.	0.217	**0.036**	**0.049**	**0.048**

Table 11: P/V Relationship VRTC Speed Shots (5/15/07)

		psi	Vel_{impact}
	3	1485	1.936
	4	1488	1.955
	5	1495	1.975
St. Dev.		**5.132**	**0.020**
	7	2480	3.514
	8	2484	3.519
	9	2493	3.515
St. Dev.		**6.658**	**0.003**
	12	3448	4.999
	13	3483	5.041
	14	3484	5.007
St. Dev.		**20.502**	**0.022**
	15	3985	5.742
	16	3988	5.745
	17	3988	5.720
St. Dev.		**1.732**	**0.014**

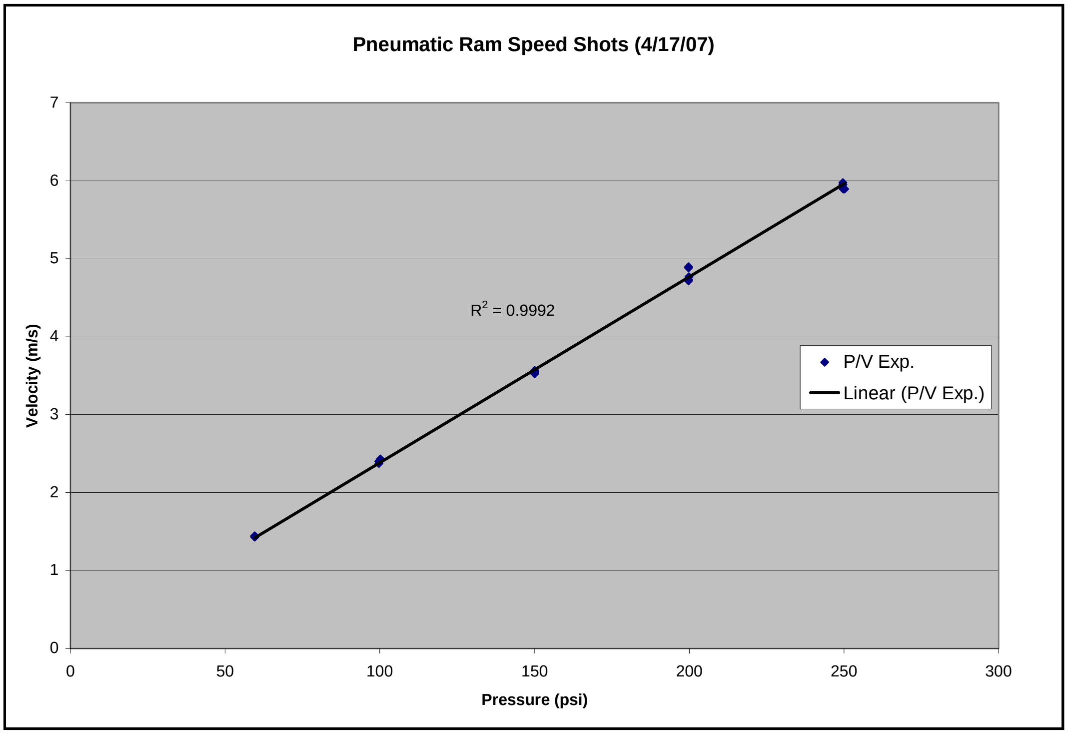


Figure 23: Pneumatic Ram Pressure/Velocity Relationship

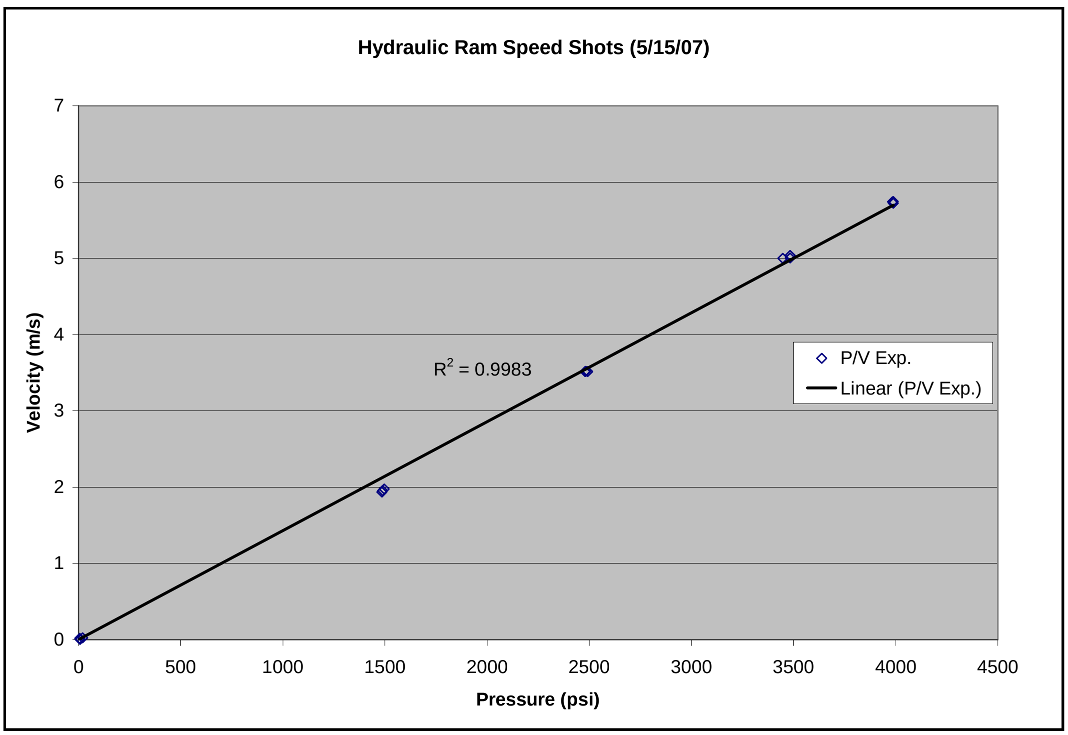


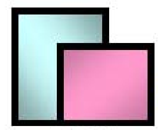
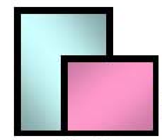
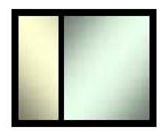
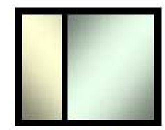
Figure 24: Hydraulic Ram Pressure/Velocity Relationship

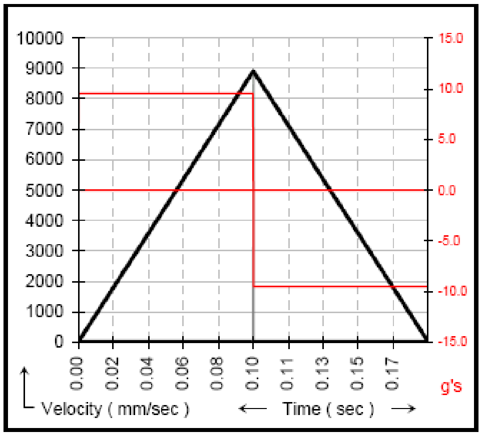
B. Linear Motor Analysis

Note: Many system variables can affect the accuracy of these estimates.
Actual performance may therefore differ from these calculated results.

Customer : Ohio State University
Project : Electromagnetic Impactor
Prepared by : David P. Broich for Chad Broering
Model Description : LZ100-HT-480DN Model

Parameters	Units	Case #1	Case #2
Stage	(Model)	X-AXIS	X-AXIS
Linear Motor	(Model)	LZ100-HT-480DN	LZ100-HT-480DN
"Move" Distance	(mm)	850	850
Anticipated Velocity Limits	(mm/sec)	10000	10000
Applied Payload	(Kg)	40.0	30.0
"S-Curve" Smooth Factor (*programmed*)	(ms)		10.0
"Dwell" Time Between Repetitive Moves	(sec)	10.000	10.000
Acceleration Current (rms)	(Arms)	34.00	34.00
Max Velocity "Achieved" (V_{max})	(mm/sec)	8920.7	9530.0
Ambient Temperature	(°C)	25	25
Acceleration (peak)	(g's)	9.547	12.271
Acceleration (average)	(g's)	9.547	10.895
Deceleration (average)	(g's)	9.547	10.895
Acceleration Dist.	(mm)	425.00	425.00
Distance @ V_{max}	(mm)		
Deceleration Dist.	(mm)	425.00	425.00
Acceleration Time	(sec)	0.0953	0.0892
Time @ V_{max}	(sec)		
Deceleration Time	(sec)	0.0953	0.0892
Move Time (including est. Settling Time)	(sec)	0.1906	0.1784
Motor's Coil Weight	(Kg)	5.04	5.04
Total Moving Weight	(Kg)	45.04	35.04
Motor's Magnetic Preload (attraction)	(N)	none	none
Max Thrust Force Calculated {@ Max Current $I_{(rms)}$}	(N)	4227	4227
Frictional Coefficient	(μ_k)	0.001	0.001
Cable Drag/Preload	(N)	8.9	8.9
Calculated Frictional Load	(N)	9.3	9.2
Translational Kinetic Energy @ V_{max} (E_k)	(Joule)	1792.31	1591.40
Dissipation Efficiency of Coil Mounting Interface	(%)	100	100
Motor Temp. @ $I_{(rms)}$	(°C)	64.4	55.4
Coil Temperature Rise above Ambient	(°C)	39.4	30.4
Current (rms) for One Move + Settle + Dwell	(amps_pk)	6.56	5.86
Required Peak Current @ Driver	(amps_pk)	48.08	48.08
Bus Votage for Achieved Velocity (V_{max})	(vdc)	1331	1402

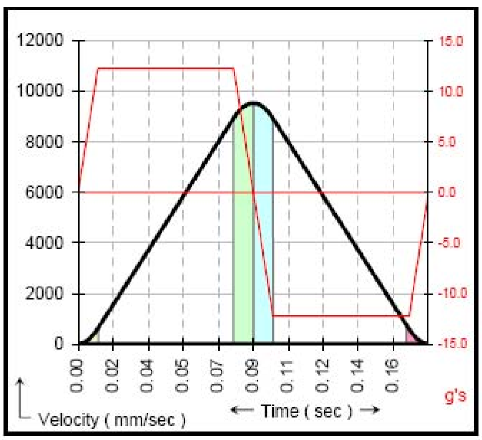
Parameters	Units	Case #1	Case #2
Stage	(Model)	X-AXIS	X-AXIS
Linear Motor	(Model)	LZ100-HT-480DN	LZ100-HT-480DN
Continuous Current Assessment Pink is % used of Available Ic (rms @ Temp Limit) coil temp	Pink = Used Blue = Icont	 64 68.2%	 55 60.9%
Peak Current Assessment Green is % of Available Ip (rms for 1-sec) used	Grn = Used Yel = Isat	 100.0%	 100.0%
Kf of Selected Motor	(N/Arms)	124.31	124.31
Ke of Selected Motor	(v/mm/sec)	0.11	0.11
R of Motor (p-p) @ 20 deg C	(ohms)	3.77	3.77
R of Motor (p-p) @ Temp.	(ohms)	4.36	4.22
L of Selected Motor (p-p)	(mh)	7.1	7.1
Rth of Selected Motor	(°C/W)	0.280	0.280
Ic of Selected Motor (rms @ max temp limit)	(Arms)	6.80	6.80
Ip of Selected Motor (rms for 1-sec)	(Arms)	34.00	34.00
Saturation Current	(Arms)	34.00	34.00
Max Velocity "Achieved" (V_{max})	(mm/sec)	8920.7	9530.0
Current (rms) During CV (@ V_{max})	(Arms)	0.08	0.07
Current During Deceleration	(Arms)	33.92	33.93
Current (rms) for One Move + Settle + Dwell	(Arms)	4.639	4.142
Current (rms) for One Move + Settle + Dwell	(amps_pk)	6.561	5.858
Electrical Cycle: E_c	(mm)	60.00	60.00
Voltage due to Back emf: K_e*V_p @V_{prog}	(vdc)	1057.15	1057.15
Voltage due to R*I: 1.225*R_{hot}*I	(vdc)	181.39	175.84
Voltage due to Inductance: 7.695*V_p*L*I/E_c @ V_{prog}	(vdc)	308.29	308.29
Bus Voltage Vdc: @ $V_{programmed}$	(vdc)	1468	1462
Voltage due to Back emf: K_e*V_p @V_{max}	(vdc)	943.05	1007.46
Voltage due to R*I: 1.225*R_{hot}*I	(vdc)	181.39	175.84
Voltage due to Inductance: 7.695*V_p*L*I/E_c @ V_{max}	(vdc)	275.01	293.80
Bus Voltage Vdc: @ V_{max}	(vdc)	1331.22	1402.11



Case #1 X-AXIS

Motor Model	LZ100-HT-480DN	Dwell Time (sec)	10.000
Moving Load (Kg)	40.0	Accel Distance (mm)	425.00
Move Distance (mm)	850.0	Dist. @ Velocity (mm)	0.00
Smooth Times (ms)	0	Decel Distance (mm)	425.00
Peak Accel (g's)	9.55	Accel Time (sec)	0.095
Average Accel (g's)	9.55	Time @ Velocity (sec)	0.000
Max Velocity (mm/s)	8920.7	Decel Time (sec)	0.095
Motor Coil Temp (°C)	64.4	Total Move Time (sec)	0.191

Note: Actual performance may differ from these calculated estimates.



Case #2 X-AXIS

Motor Model	LZ100-HT-480DN	Dwell Time (sec)	10.000
Moving Load (Kg)	30.0	Accel Distance (mm)	425.00
Move Distance (mm)	850.0	Dist. @ Velocity (mm)	0.00
Smooth Times (ms)	10	Decel Distance (mm)	425.00
Peak Accel (g's)	12.27	Accel Time (sec)	0.089
Average Accel (g's)	10.90	Time @ Velocity (sec)	0.000
Max Velocity (mm/s)	9530.0	Decel Time (sec)	0.089
Motor Coil Temp (°C)	55.4	Total Move Time (sec)	0.178

Note: Actual performance may differ from these calculated estimates.

C. Air Bearing/Shaft Quote

MOTIONUSA.com
Positioning Solutions

Motion Technologies Co.
1205 Chesapeake Avenue
Columbus, Ohio 43212
Voice: 614-487-1417
Fax: 614-487-8606
Mobile: 614-496-4587
E-Mail: sales@motionusa.com

October 31, 2008

Chad Broering
Ohio State University
Injury Biomechanics Research Lab
3024 Graves Hall
333 West Tenth Avenue
Columbus, OH 43210

broering.24@gmail.com
Ph: (419) 733-9585

Ref: **New Way Air Bushings**
MTC QUOTE NO: 81031-JN1

Chad,

Thank you for the opportunity to quote your requirements. The quoted units are the New Way air bushings and aluminum shafts that we have been working on for your accelerator project.

Please feel free to contact me with any questions. Motion Technologies thanks you for the opportunity to quote your requirements. We look forward to assisting the Injury Biomechanics Research Lab on this project and with your future opportunities.

Sincerely,

John Nemergut

Motion Technologies Co.

MOTIONUSA.com
Positioning Solutions

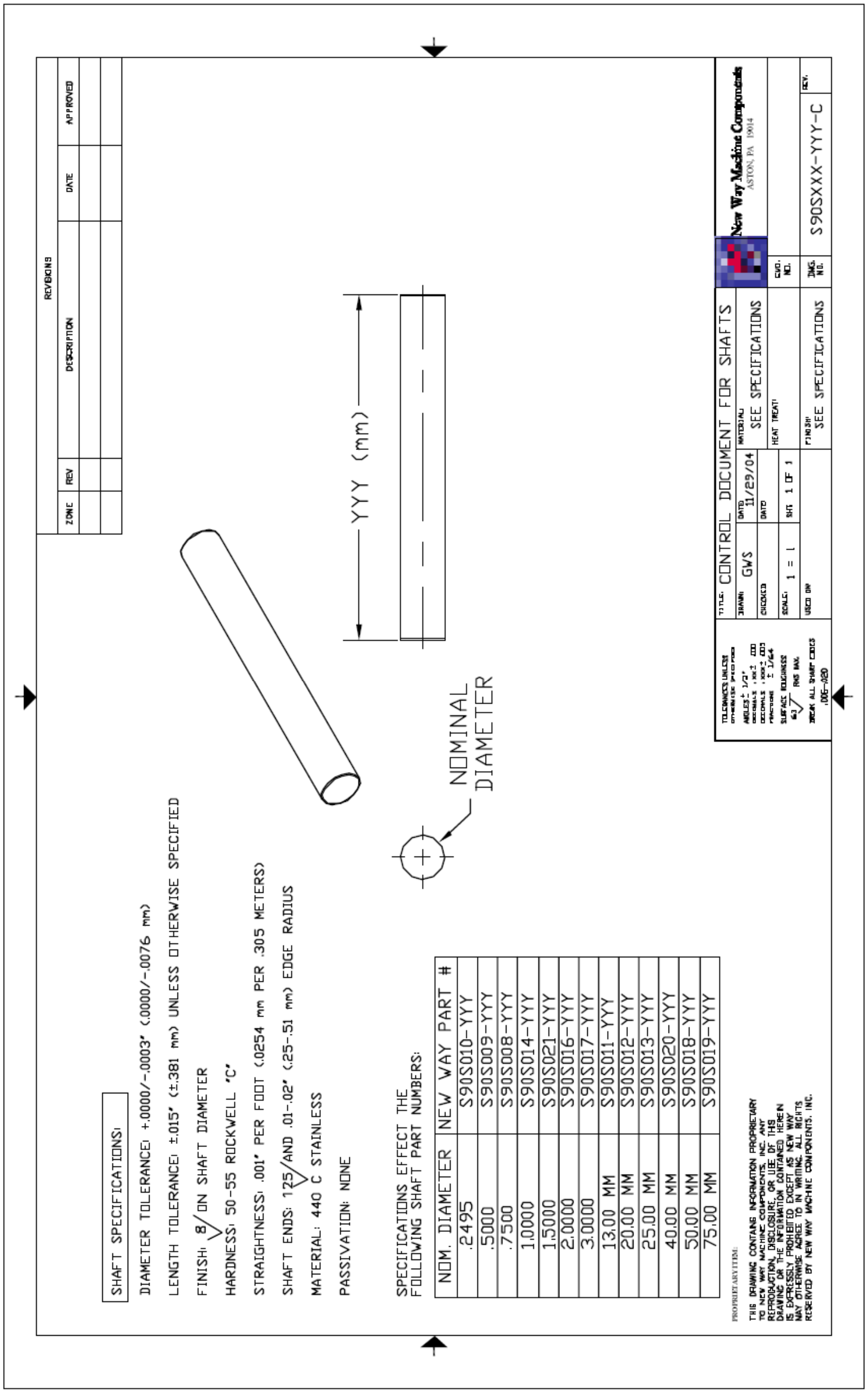
Motion Technologies Co.
1205 Chesapeake Avenue
Columbus, Ohio 43212
Voice: 614-487-1417
Fax: 614-487-8606
Mobile: 614-496-4587
E-Mail: sales@motionusa.com

Chad Broering
Ohio State University
Injury Biomechanics Research Lab
Ref: **New Way Air Bushings**
MTC QUOTE NO: 81031-JN1
October 31, 2008

Item	Part Number & Description	Qty	Price Ea	Ext. Price
1	**P/N: S305001** *New Way Air Bushing* ▪ 2.00" I.D. air bushing	4	$420.00	$1,680.00
2	**P/N: S8050P02** *New Way Mounting Block* ▪ Mounting block for 2.00" I.D. air bushing	4	$230.00	$920.00
3	**P/N: S90S016-AL-30IN** *Aluminum Shaft* ▪ Solid aluminum shaft ▪ Tolerance for air bushings ▪ 30in length	2	$495.00	$990.00
			TOTAL:	**$3,590.00**

Terms & Conditions	
Order Placement:	Address Purchase Order to Motion Technologies Co.; 1205 Chesapeake Ave.; Columbus, OH 43212
F.O.B.:	MFG factories.
Delivery:	Standard delivery is about 2-4 weeks after receipt of order.
Payment Terms:	Net30 days after date of invoice.
Validity:	Quote is valid for 30 days.

D. Shaft Specifications



E. Alro Quote

ALRO STEEL CORPORATION

QUOTATION

555 ROME HILLIARD
COLUMBUS, OHIO 43228 614 878-7271
DUNS: 15-764-2674

ALRO QUOTE: 037160226
DATE: 10/29/08
CUST ORD:
CUST REL:

ATTN : CHAD BEOERING
ACCT# : 00036910
COMPANY : **Ohio State University**
ADDRESS : Attn: Accounts Payable
Columbus, OH 43210
614 292-9822

SHIP TO: Ohio State University
Attn: Accoutns Payable
901 Woody Hayes Drive
Columbus, OH 43210

FOB: YOUR PLANT
SHIP VIA: CB TRUCK

FROM: GINNIE SMITH
LOCATION: COLUMBUS
FAX#: 614-878-0769

ASK US ABOUT DC53...A GREAT ALTERNATIVE TO D2/A2/M2!

LINE	ORD QTY	UNIT	PART#	ITEM DESCRIPTION		WEIGHT	PRICE		EXT.PRICE	DELIVERY DATE
1	1.00	PC	07008805	1 A-36 PLATE 38 X 76 IN Cut Tolerance: +1/8 / -0	MISC		899.0000	PC	899.00	
2	1.00	PC	13035699	8 X 2 X 3/8 WALL TUBING 48 IN ASTM A-500 GRADE B MISC Cut Tolerance: +1/8 / -0			126.7500	PC	126.75	
3	1.00	PC	04536899	4 X 3 X 3/8 A-36 ANGLE 96 IN Cut Tolerance: +1/8 / -0			84.2500	PC	84.25	
4	2.00	LNG	04507520	2-1/2 X 2-1/2 X 1/4 A-36 ANGLE 20 FT		164.00	77.5000	CWT	127.10	
5	2.00	PC	07007205	3/4 A-36 PLATE CP# PER PRINT 36 X 76 Cut Tolerance: +1/8 / -0 PER PRINT	MISC		609.7500	PC	1219.50	
6	2.00	PC	07007205	3/4 A-36 PLATE 12 1/2 X 30 IN Cut Tolerance: +1/8 / -0	MISC		95.2500	PC	190.50	
7	2.00	PC	06511120	1/2 X 12 A-36 HR STEEL 24 IN Cut Tolerance: +1/8 / -0			46.2500	PC	92.50	

CONTINUED ON NEXT PAGE

ALRO STEEL CORPORATION

555 ROME HILLIARD
COLUMBUS, OHIO 43228   614 878-7271
DUNS: 15-764-2674

QUOTATION

ALRO QUOTE: 037160226
DATE: 10/29/08
CUST ORD:
CUST REL:

ATTN : CHAD BEOERING
ACCT# : 00036910
COMPANY : **Ohio State University**
ADDRESS : Attn: Accounts Payable
Columbus, OH 43210
614 292-9822

SHIP TO: Ohio State University
Attn: Accoutns Payable
901 Woody Hayes Drive
Columbus, OH 43210

FOB: YOUR PLANT
SHIP VIA: CB TRUCK

FROM: GINNIE SMITH
LOCATION: COLUMBUS
FAX#: 614-878-0769

ASK US ABOUT DC53...A GREAT ALTERNATIVE TO D2/A2/M2!

LINE	ORD QTY	UNIT	PART#	ITEM DESCRIPTION	WEIGHT	PRICE	EXT.PRICE	DELIVERY DATE
8	1.00	PC	06510920	1/2 X 8 A-36 HR STEEL 53 IN Cut Tolerance: +1/8 / -0		71.2500 PC	71.25	

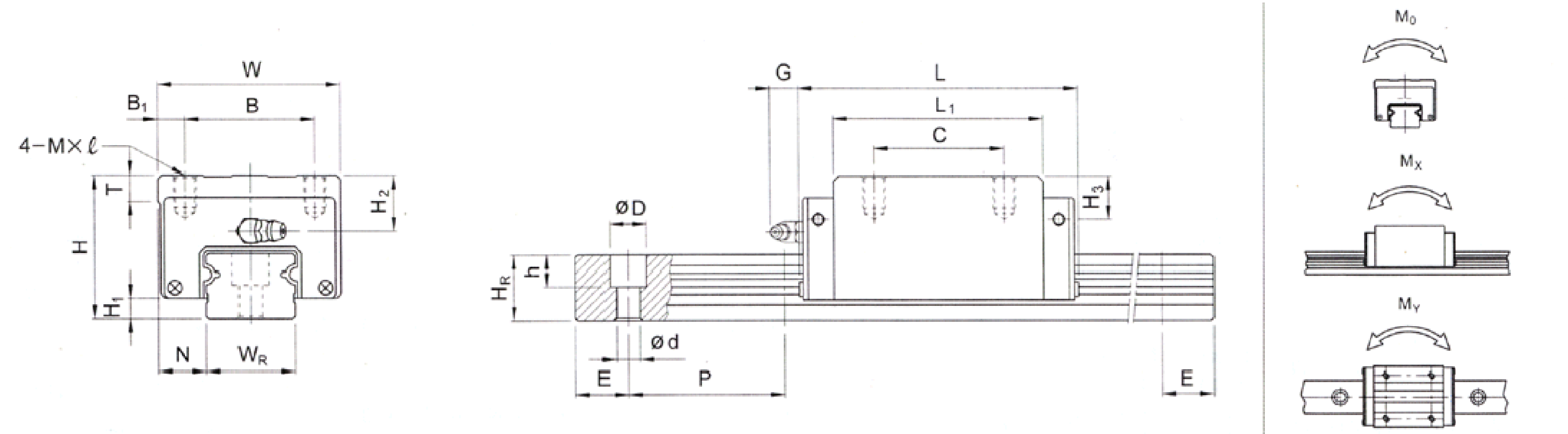
TOTAL LINES: 8 **QTY:** 12.00 **WEIGHT:** 164.00 **FUEL SURCHARGE:** 10.60 **TAX:** **TOTAL PRICE:** 2821.45

PO# ____________ **SIGNED** ____________ **FAX QUOTE ACCEPTANCE TO: 614-878-0769**

DUE DATE ____________ **DATE** ____________

* POUNDS SHOWN ARE BASED ON MATERIAL REQUIRED TO PRODUCE AND FILL YOUR ORDER AND ARE BASED ON CALCULATED WEIGHTS WITHIN NORMAL MILL TOLERANCES AND MAY VARY FROM ACTUAL WEIGHT SHIPPED.
* THE ABOVE QUOTE IS FOR YOUR INTERNAL USE ONLY AND SHOULD NOT BE SHARED WITH ANY THIRD PARTY IN ANY FORM.
* AVAILABILITY SUBJECT TO PRIOR SALE(S).
* ALL AMOUNTS ARE STATED IN U.S.DOLLARS & MUST BE PAID IN U.S.DOLLARS
* PRICES QUOTED WILL BE HONORED IF ORDERED AND SHIPPED WITHIN 48 HOURS OF THIS QUOTE.
* ALL OTHER ORDERS WILL BE PRICED AT PRICING LEVELS AT TIME OF DELIVERY.
* PRICES INCLUDE RAW MATERIAL SURCHARGES WHERE APPLICABLE.
* PRICES ARE PREDICATED ON RECEIVING THE TOTAL ORDER

F. Hiwin Product Specifications



Model No.	Dimensions of Assembly (mm)			Dimensions of Block (mm)										Dimensions of Rail (mm)							Mounting Bolt for Rail (mm)	Basic Dynamic Load Rating C (kgf)	Basic Static Load Rating C_0(kgf)	Static Rated Moment			Weight	
	H	H_1	N	W	B	B_1	C	L_1	L	G	M×ℓ	T	H_2	W_R	H_R	D	h	d	P	E				M_0 (kgf-m)	M_X (kgf-m)	M_Y (kgf-m)	Block (kg)	Rail (kg/m)
LGH 15CA	28	4.5	9.5	34	26	4	26	39.6	60.6	5.3	M4×5	6	8.5	15	14	7.5	5.3	4.5	60	20	M4×16	1,040	1,680	13.5	11.0	11.0	0.21	1.47
LGH 20CA	30	5	12	44	32	6	36	52.7	77.3	12	M5×6	8	7.1	20	15	9.5	8.5	6	60	20	M5×16	1,650	2,670	28.1	22.8	22.8	0.37	2.08
LGH 20HA							50	67	91.6													2,100	3,400	35.7	35.9	35.9	0.46	
LGH 25CA	40	6.5	12.5	48	35	6.5	35	57.6	85.6	12	M6×8	8	11.2	23	20	11	9	7	60	20	M6×20	2,410	3,880	46.6	37.2	37.2	0.59	3.15
LGH 25HA							50	76.6	104.6													3,210	5,180	62.2	63.6	63.6	0.78	
LGH 30CA	45	7	16	60	40	10	40	72	104.4	12	M8×10	8	10.5	28	23	14	12	9	80	20	M8×25	3,380	5,460	79.3	61.2	61.2	1.04	4.41
LGH 30HA							60	93	125.4													4,400	7,100	103.0	100.4	100.4	1.33	
LGH 35CA	55	8	18	70	50	10	50	82	118.4	12	M8×12	10	15	34	25	14	12	9	80	20	M8×25	4,180	6,740	118.1	84.4	84.4	1.72	5.93
LGH 35HA							72	105.8	142.2													5,430	8,770	153.5	138.4	138.4	2.24	
LGH 45CA	70	10	20.5	86	60	13	60	99.6	139.2	12.9	M10×17	15	21	45	32	20	17	14	105	22.5	M12×35	6,020	9,710	223.5	141.3	141.3	3.16	10.01
LGH 45HA							80	133	172.6													8,430	13,600	312.8	259.2	259.2	4.28	
LGH 55CA	80	13	23.5	100	75	12.5	75	115.8	164.8	12.9	M12×18	17	22	53	40	23	20	16	120	30	M14×45	9,740	13,220	384.9	280.9	280.9	5.30	14.82
LGH 55HA							95	154.7	203.7													11,810	18,510	489.8	442.7	442.7	6.40	
LGH 65CA	90	19	31.5	126	76	25	70	138.6	197.6	12.9	M16×20	25	20	63	48	26	22	18	150	35	M16×50	14,940	20,990	738.8	579.0	579.0	7.30	21.26
LGH 65HA							120	187.6	246.6													18,290	27,290	1007.5	1040.8	1040.8	9.30	

Above listed dimensions of rail are dimensions of LGR-R (Bolt hole, mounting from above), and dimensions of LGR-T (Tapped hole, mounting form below) refer to Page 20.

G. MotionUSA Quote (Vertical Test Frame)

MOTIONUSA.com
Positioning Solutions

Motion Technologies Co.
1205 Chesapeake Avenue
Columbus, Ohio 43212
Voice: 614-487-1417
Fax: 614-487-8606
Mobile: 614-496-4587
E-Mail: sales@motionusa.com

October 31, 2008

Chad Broering
Ohio State University
Injury Biomechanics Research Lab
3024 Graves Hall
333 West Tenth Avenue
Columbus, OH 43210

broering.24@gmail.com
Ph: (419) 733-9585

Ref: **Vertical Lift Assembly – Initial Pricing**
MTC QUOTE NO: 81031-JN2

Chad,

Thank you for the opportunity to quote your requirements. The quoted unit is the assembled base unit with 2 linear square rails on each side and 2 blocks on each rail. It was done up to match the assembly drawing that you provided: “Final Design (Skamar).igs”.

Please note that this is an incomplete package. Not included here are the acme screw and nut assembly, gearbox, and the hand crank and/or servo package to drive the unit up and down.

Please feel free to contact me with any questions. Motion Technologies thanks you for the opportunity to quote your requirements. We look forward to assisting the Injury Biomechanics Research Lab on this project and with your future opportunities.

Sincerely,

John Nemergut

Motion Technologies Co.

Motion Technologies Co.
1205 Chesapeake Avenue
Columbus, Ohio 43212
Voice: 614-487-1417
Fax: 614-487-8606
Mobile: 614-496-4587
E-Mail: sales@motionusa.com

Chad Broering
Ohio State University
Injury Biomechanics Research Lab
Ref: **Vertical Lift Assembly – Initial Pricing**
MTC QUOTE NO: 81031-JN2
October 31, 2008
Page 2

Item	Part Number & Description	Price

1	**P/N: SKA-FRM-81031**
	Skamar Frame Assembly for Vertical Stage
	▪ Frame for vertical stage as shown in concept drawing
	▪ Includes:
	  o Parts for weldment assembly
	  o Welding of assembly
	  o Machining of assembly
	  o 4 linear rails
	  o 8 bearings for linear rails
	  o 4 manual clamps for linear rails
	  o Assembly of parts
	▪ This structure does not yet include the price for the drivetrain
		$12,500.00

Terms & Conditions	
Order Placement:	Address Purchase Order to Motion Technologies Co.; 1205 Chesapeake Ave.; Columbus, OH 43212
F.O.B.:	Skamar Machine: Cleveland, OH.
Delivery:	Delivery to be determined before order placement.
Payment Terms:	Net30 days after date of invoice.
Validity:	Quote is valid for 30 days.

H. Pin Shearing

$$\tau_s = \frac{4F_s}{\pi D_p^2} = \frac{4\left(\frac{35{,}500}{2}\right)}{\pi * 0.03175^2} = 22.42\ N/m^2$$

Assuming AISI No. C 1018 cold drawn steel, Yield Point=441.264 MPa.

$$S.F. = \frac{YP}{\tau_s} = \frac{441.264}{22.42} = 19.68$$

This value is comparable to the literature value from the Pin Joint section.

I. MATLAB Code

```
%----------------------------------------------------------------------%
%Written by: Chad Broering
%This code calculates the force requirements of an accelerator assuming
%constant velocity, and zero friction.  These calculations are based solely
%on inertial forces.
%Units are SI unless otherwise specified.
%----------------------------------------------------------------------%

clc; clear all; close all;

%----------Part 1----------%
%mmax=maximum ram weight
%vmax=maximum final velocity
%laccr=range of acceleration phase lengths
%a=calculated acceleration
%Fmax=required force at various acceleration ranges and maximum mass
mmax=23.5;
vmax=10;
laccr=(.1:.01:.6);
a=(vmax^2)./(2*laccr);
Fmax=mmax*a;

figure(1)
plot(laccr,Fmax);  title('F_m_a_x vs. Desired Acceleration Length (v_m_a_x =
10 m/s, Mass=23.5 kg)');
xlabel('Length of Acceleration Phase (m)');
ylabel('Force Required (N)'); grid on;

%----------Part 2----------%
%m=applicable range of ram masses
%v=range of desired velocities
%lacc=acceleration lengths
%a=calculated acceleration
%F=force matrix (Rows=reqiured forces, Columns=discrete mass values)
m=[10:2.5:37.5]';
v=[1:.1:10];
lacc=.30;
a=(v.^2)./(2*lacc);
  for i=1:length(m)
     F(i,:)=m(i).*a;
  end

figure(2)
plot(v,F);  title('Force vs. Velocity for Varying Mass Weights (30 cm accel
phase)');
ylabel('Force (N)');  xlabel('Velocity (m/s)'); grid on;
legend('10 kg','12.5 kg','15 kg','17.5 kg','20 kg','22.5 kg','25 kg','27.5
kg','30 kg','32.5 kg','35 kg','37.5 kg')
```

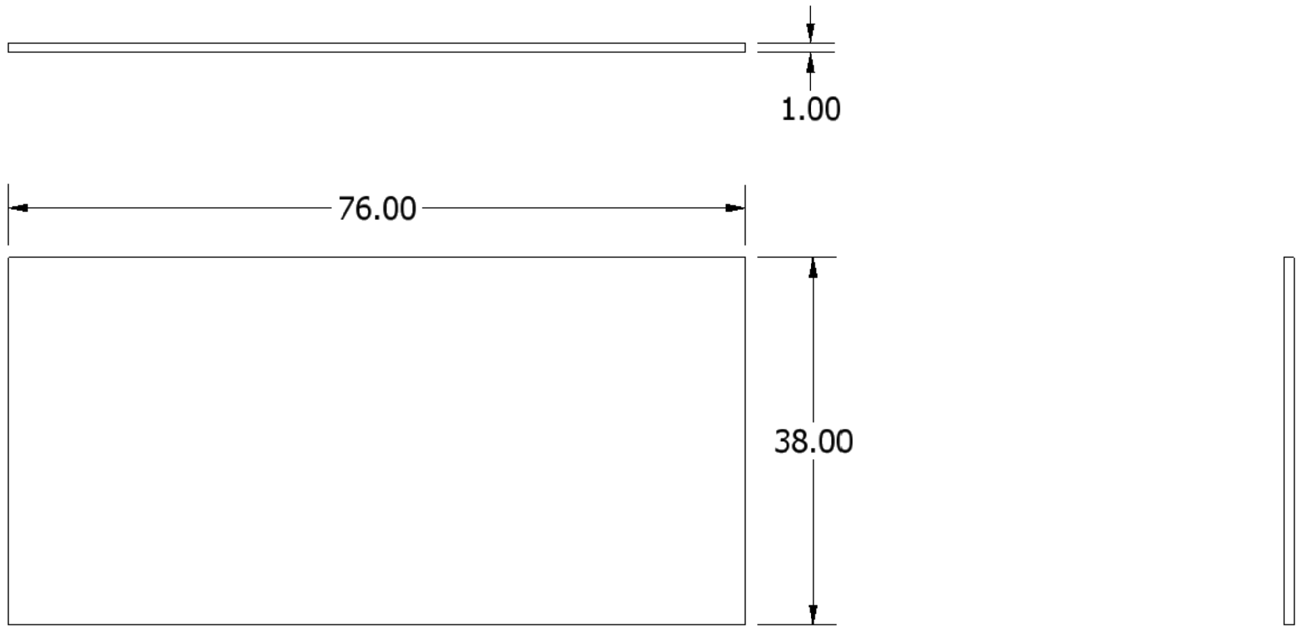
```
%----------------------------------------------------------------------%
%Written by: Chad Broering
```

```matlab
%This code calculates the cylinder velocity based on the ratio of the area
%of the throat to the area of the cylinder.
%DtDc: Ratio of the diameter of the throat divided by the diameter of
%the cylinder.
%DelP: Accumulator pressure-vapor pressure (psi)
%Vc: Cylinder velocity (.3048 converts ft/s to m/s)
%-------------------------------------------------------------------------%
clc; clear all; close all;

Cd=.98;
DtDc=[.1:.1:.7];
delP=[0:1:3000];

for i=1:length(DtDc)
    Vc(i,:)=(DtDc(i)^2*Cd*5.282*delP.^.5)*.3048;
end

plot(Vc(:,:),delP)
title('Pressure vs Velocity Curves for a Cavitating Venturi (Discrete Values
of D_t_h_r_o_a_t/D_c_y_l_i_n_d_e_r)');
xlabel('Acuator Velocity (m/s)'); legend('.1','.2','.3','.4','.5','.6','.7');
ylabel('Available deltaP (psi) [deltaP = Inlet Pressure - Vapor Pressure]');
```

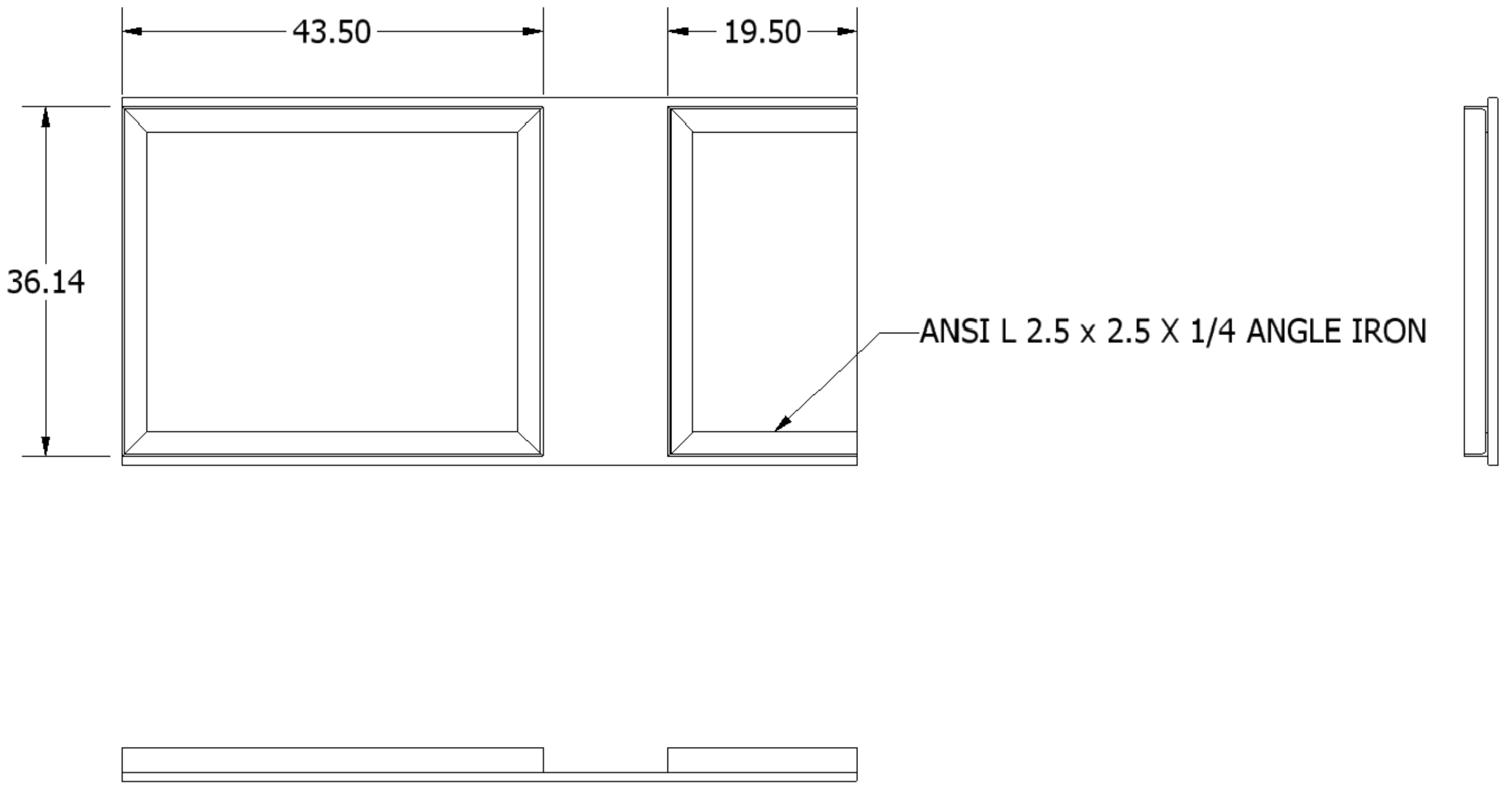
```
%-------------------------------------------------------------------------%
%Written by: Chad Broering
%This codes calculates the maximum downforces at the front face of the ram
%based on bearing performance criteria and a moment analysis assuming
%lengths of current ram.
%Reference Notebook pgs 48 & 22
%-------------------------------------------------------------------------%

clc; close all; clear all; format compact;

%----------Constants----------%
%Fb2max=Maximum bearable radial force (2 x 2.0" bearings @ 80 psi)
%Fr=Radial Force on Bearing (Downward) N
%W=Bearing Mass/2 (Assumed that masses are located at ends) N
%xb=Distance between bearings (edge to edge) m
%xramfe=Distance from rear bearing (edge) to front end of ram
(extended,center of
    %face mass) m (pg 66 in notebook)
%xrambe=Distance from rear bearing (edge) to rear end of ram (extended,
center of
    %face mass) m
Fb2max=2250;
Fr=0;
W=0;
xb=.2540;
xramfe=.8033;
xrambe=.0825;

%----------Frmax for varying mass extended position----------%
%Wa=ram weights
%Frmax=max force that can be applied at ram front face
Wa=[10:.1:37.5];       %kg
W=Wa*9.805;            %N
Frmax=(W*xrambe+Fb2max*xb-W*xramfe)/(xramfe);  %N
Frmaxkg=Frmax/9.805;                           %kg
Frmaxlb=Frmaxkg*2.2;                           %lb
figure(1)
plot(Wa,Frmaxlb); grid on; title('Downforce_m_a_x (lb) for Varying Ram Masses
at Extended Position');
xlabel('Mass of Ram (kg)'); ylabel('Downforce_m_a_x at Front End of Ram
(lb)');

%----------Frmax for 10, 23.5, and 37.5 kg and varying positions----------%
%xext=Inverse Position of ram m (length from completely extended) m
%xext=Position of ram (center to full extended) m
%xramfe1=Distance from rear bearing (edge) to front end of ram m
%xrambe1=Distance from rear bearing (edge) to rear end of ram m
%W##=Bearing Mass/2 (Assumed that masses are located at ends) N
%Frmax###=max force that can be applied at ram front face
xext=[0:.0001:.3604];
xext1=.3604-xext+.4429;
xramfe1=xramfe-xext;
xrambe1=xrambe+xext;
W10=10*9.805;
W235=23.5*9.805;
W375=37.5*9.805;
```

```
Frmax10=((W10*xrambe1+Fb2max*xb-W10*xramfe1)./(xramfe1))*.2248;
F10=max(Frmax10)-Frmax10;
Frmax235=((W235*xrambe1+Fb2max*xb-W235*xramfe1)./(xramfe1))*.2248;
F235=max(Frmax235)-Frmax235;
Frmax375=((W375*xrambe1+Fb2max*xb-W375*xramfe1)./(xramfe1))*.2248;
F375=max(Frmax375)-Frmax375;
figure(2)
plot(xext1,Frmax10,xext1,Frmax235,'--',xext1,Frmax375,'-.'); grid on;
title('Downforce_m_a_x (lb) for Discrete Ram Mass Values at Varying
Positions');
xlabel('Length Ram is Extended, "cm"'); ylabel('Downforce_m_a_x at Front End
of Ram (lb)');
legend('10 kg Mass','23.5 kg Mass', '37.5 kg Mass');

%----------Frmax for 37.5 kg mass at extended position for varying bearing
widths----------%
%xb=Distance between bearings (edge to edge) m
xb=linspace(.2540,.1651,100);
xbin=xb/.0254;
W=37.5*9.805;
xramfe2(1)=xramfe;
xrambe2(1)=xrambe;
for i=1:99
xramfe2(i+1)=xramfe2(i)-(.2540-.1651)/100;
xrambe2(i+1)=xrambe2(i)+(.2540-.1651)/100;
end
Frmax=(W*xrambe2+Fb2max*xb-W*xramfe2)./(xramfe2);   %N
Frmaxkg=Frmax/9.805;                               %kg
Frmaxlb=Frmaxkg*2.2;                               %lb
figure(3)
plot(xbin,Frmaxlb); grid on; title('Downforce_m_a_x (lb) for 37.5 kg Ram Mass
for Varying Bearing Widths (in)');
xlabel('Bearing Length (in)'); ylabel('Downforce_m_a_x at Front End of Ram
(lb)');
```

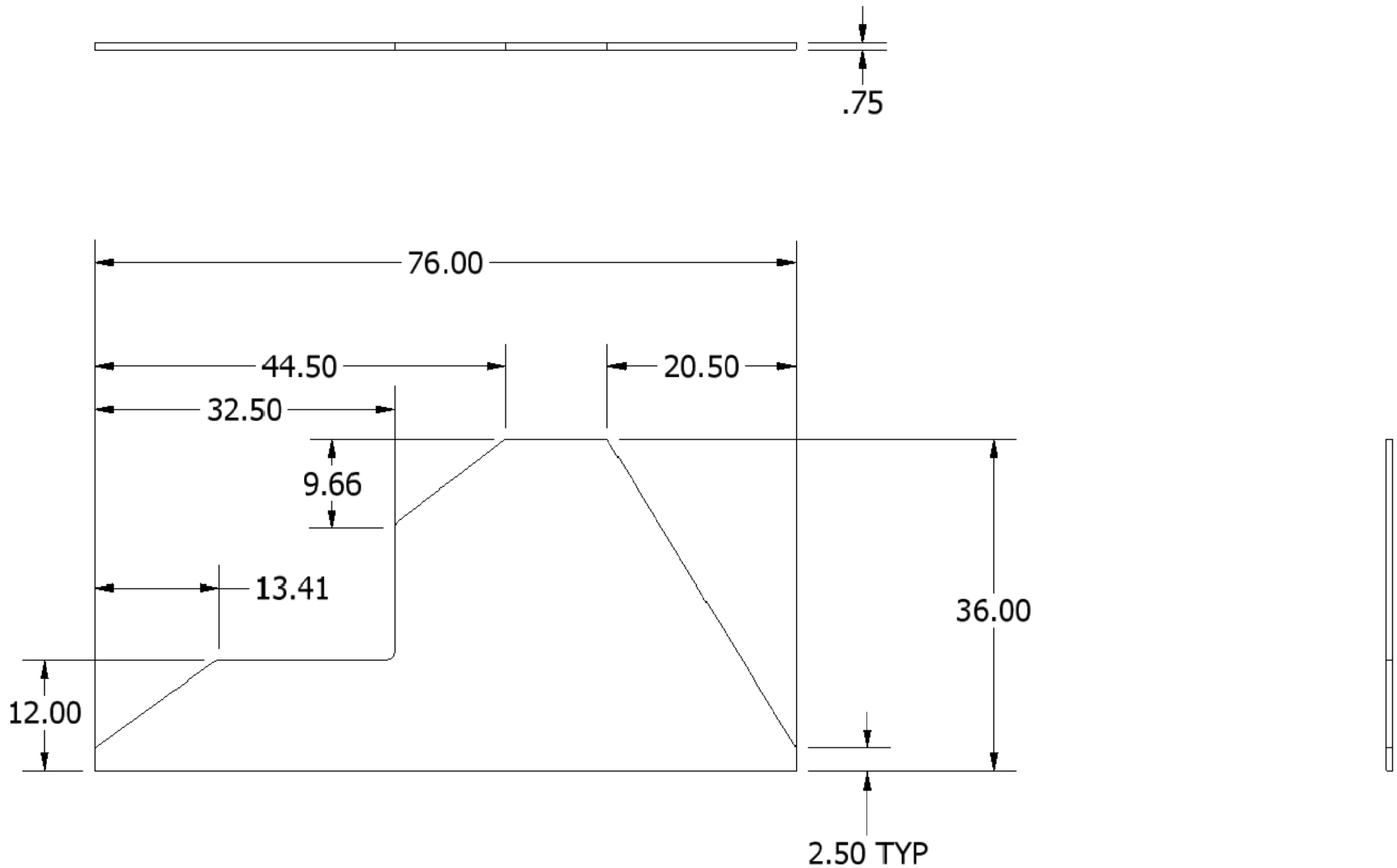
J. Drawings



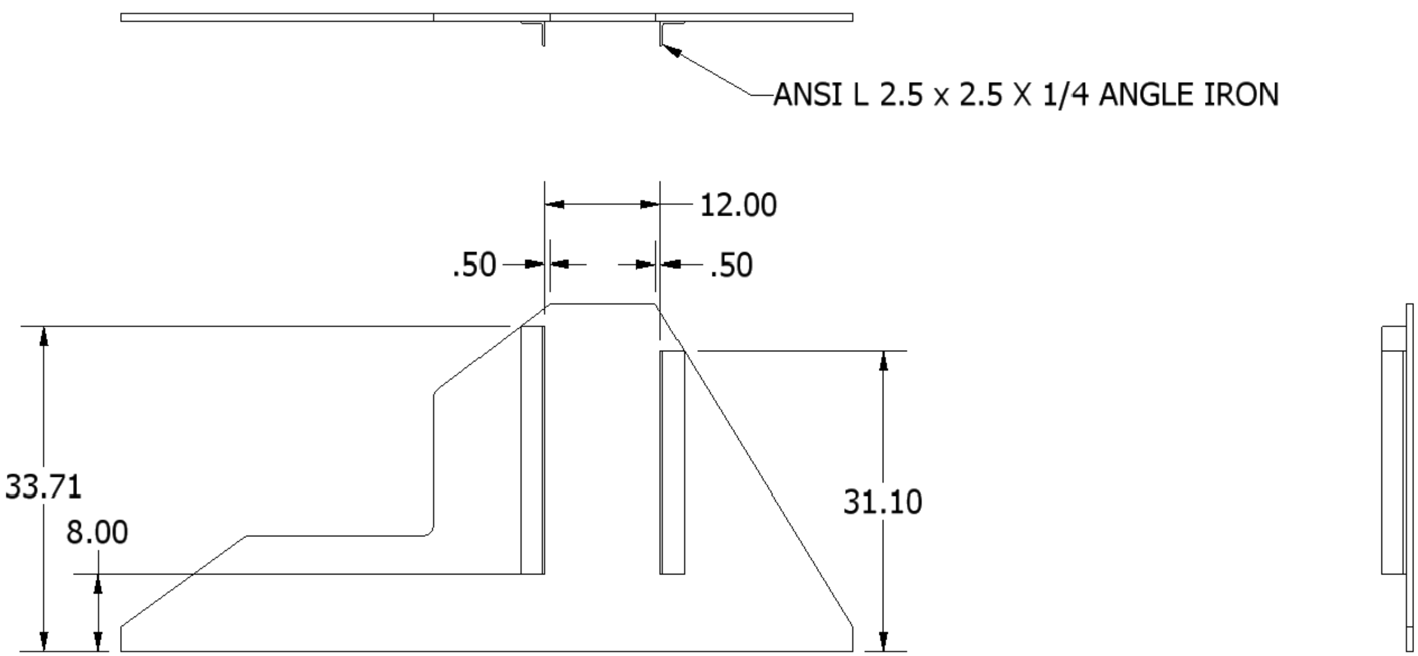
DRAWN: CHAD BROERING		INJURY BIOMECHANICS RESEARCH LAB
MATERIAL: A-36 HRS		TITLE: BASE PLATE
ASSEMBLY: STATIC SUPPORT STRUCTURE		DESCRIPTION: 76"X38"X1" BASE PLATE
DATE: 10/31/2008		
SHEET SIZE: A	SCALE	NOTES: DRILLED AND TAPPED HOLE LOCATIONS NOT SPECIFIED
CHECKED BY:	1:20	SHEET 1 OF 26



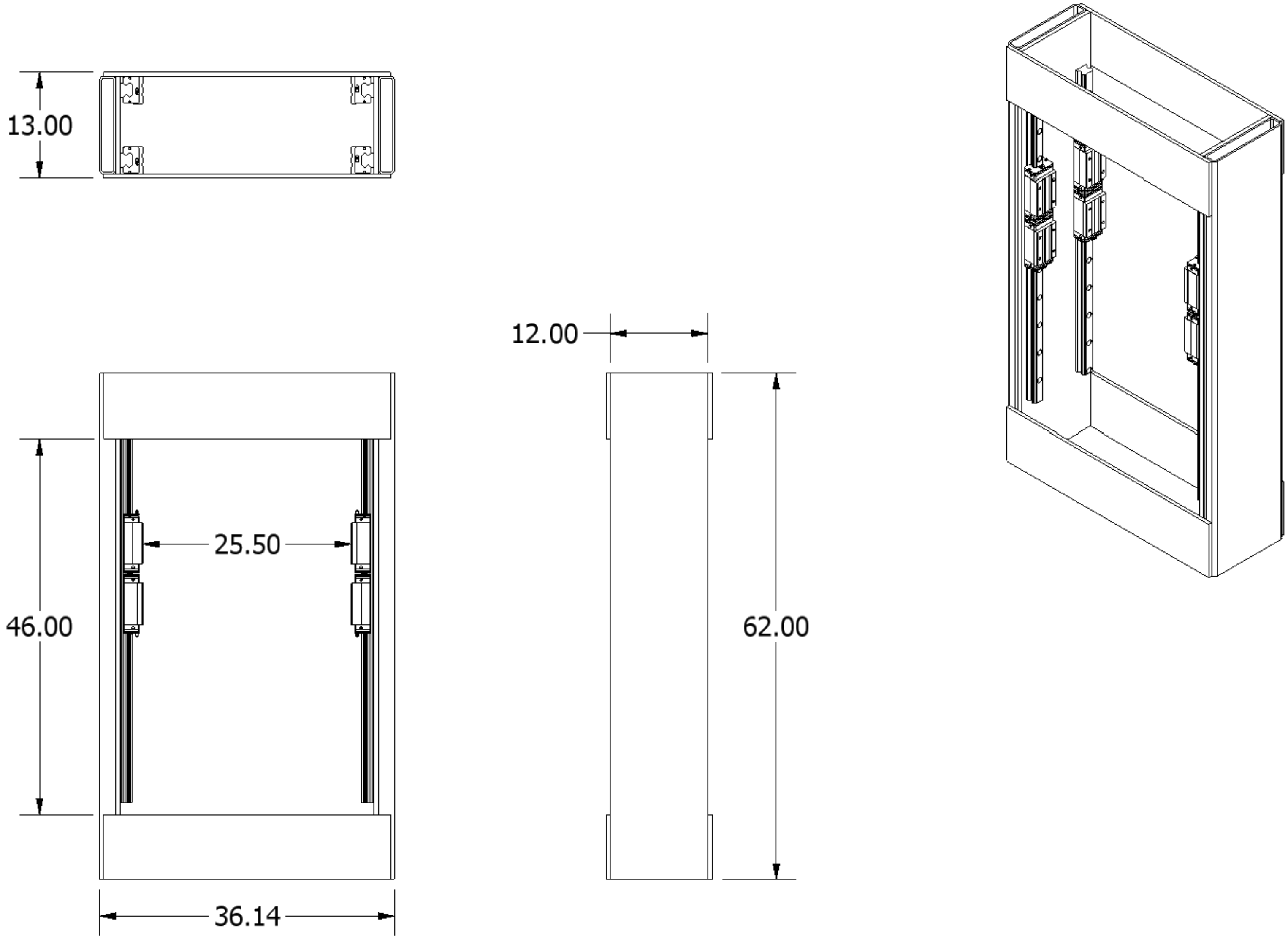
DRAWN: CHAD BROERING	INJURY BIOMECHANICS RESEARCH LAB
MATERIAL: A-36 HRS	TITLE: BASE PLATE ANGLE IRON
ASSEMBLY: STATIC SUPPORT STRUCTURE	DESCRIPTION: ANGLE IRON LOCATIONS ARE SPECIFIED IN THIS DRAWING
DATE: 10/31/2008	
SHEET SIZE: A / SCALE 1:20	NOTES: DRILLED AND TAPPED HOLE LOCATIONS NOT SPECIFIED
CHECKED BY:	SHEET 2 OF 26



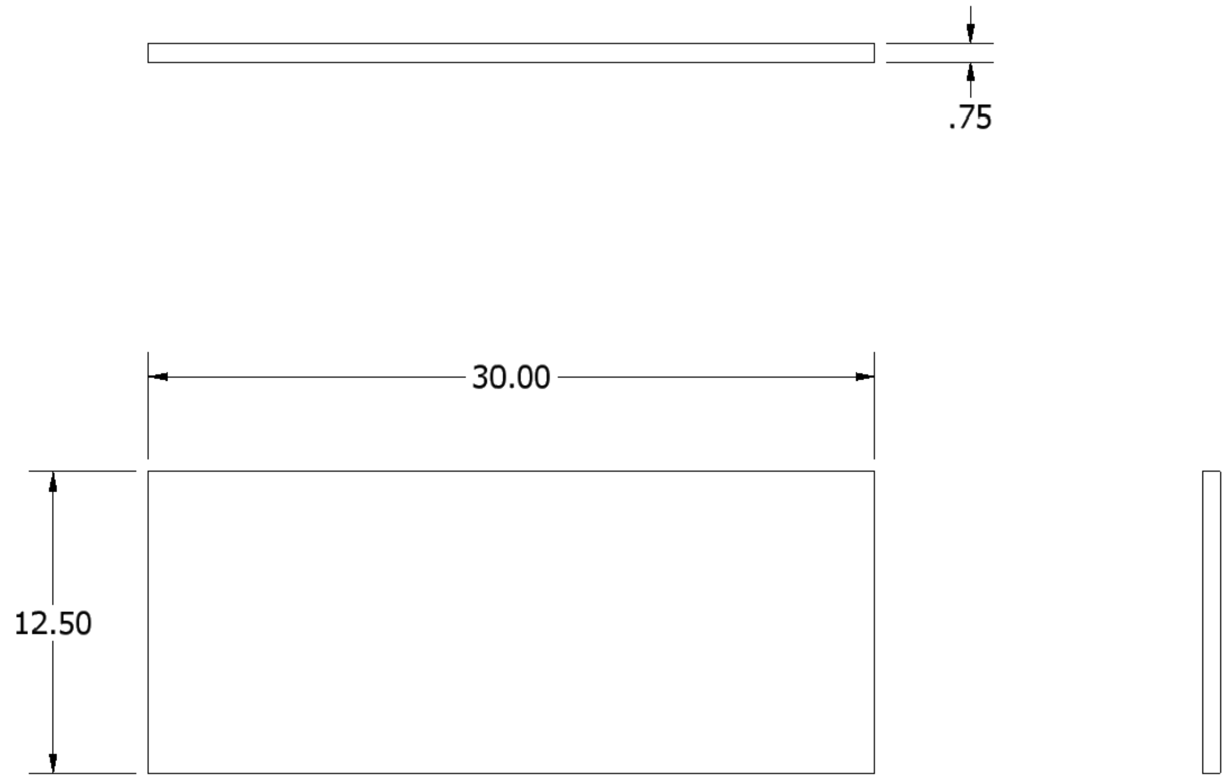
DRAWN: CHAD BROERING		INJURY BIOMECHANICS RESEARCH LAB
MATERIAL: A-36 HRS		TITLE: GUSSET
ASSEMBLY: STATIC SUPPORT STRUCTURE		DESCRIPTION: SUPPORT GUSSETS FOR TEST FRAME
DATE: 10/31/2008		
SHEET SIZE: A	SCALE	NOTES: DRILLED AND TAPPED HOLE LOCATIONS NOT SPECIFIED
CHECKED BY:	1:20	SHEET 3 OF 26



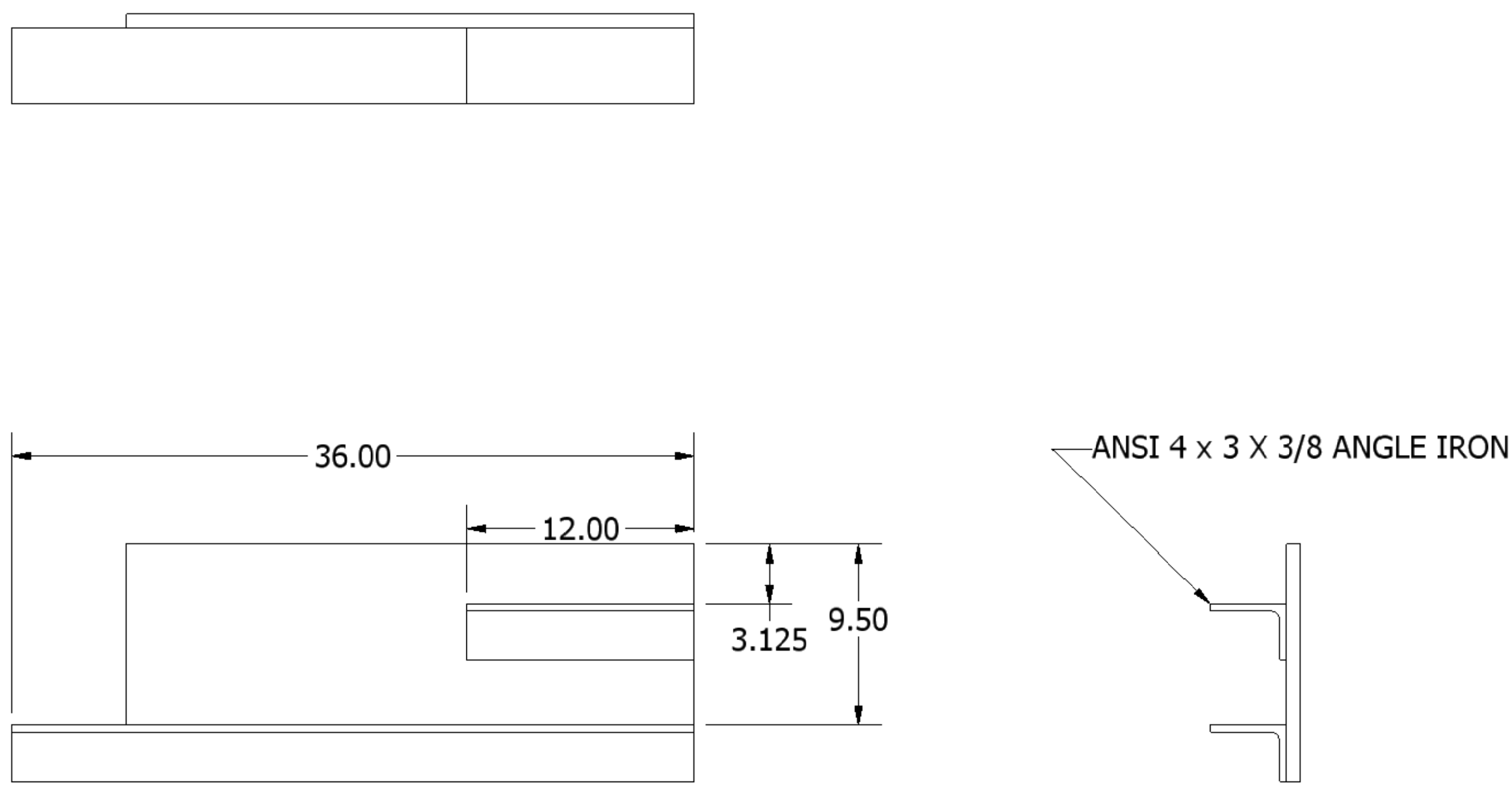
DRAWN: CHAD BROERING		INJURY BIOMECHANICS RESEARCH LAB	
MATERIAL: A-36 HRS		TITLE: GUSSET ANGLE IRON	
ASSEMBLY: STATIC SUPPORT STRUCTURE		DESCRIPTION: GUSSET WITH ANGLE IRON LOCATIONS SPECIFIED	
DATE: 10/31/2008			
SHEET SIZE: A	SCALE	NOTES: HOLE LOCATIONS NOT SHOWN. 2 PCS (MIRROR IMAGES).	
CHECKED BY:	1:20		SHEET 4 OF 26



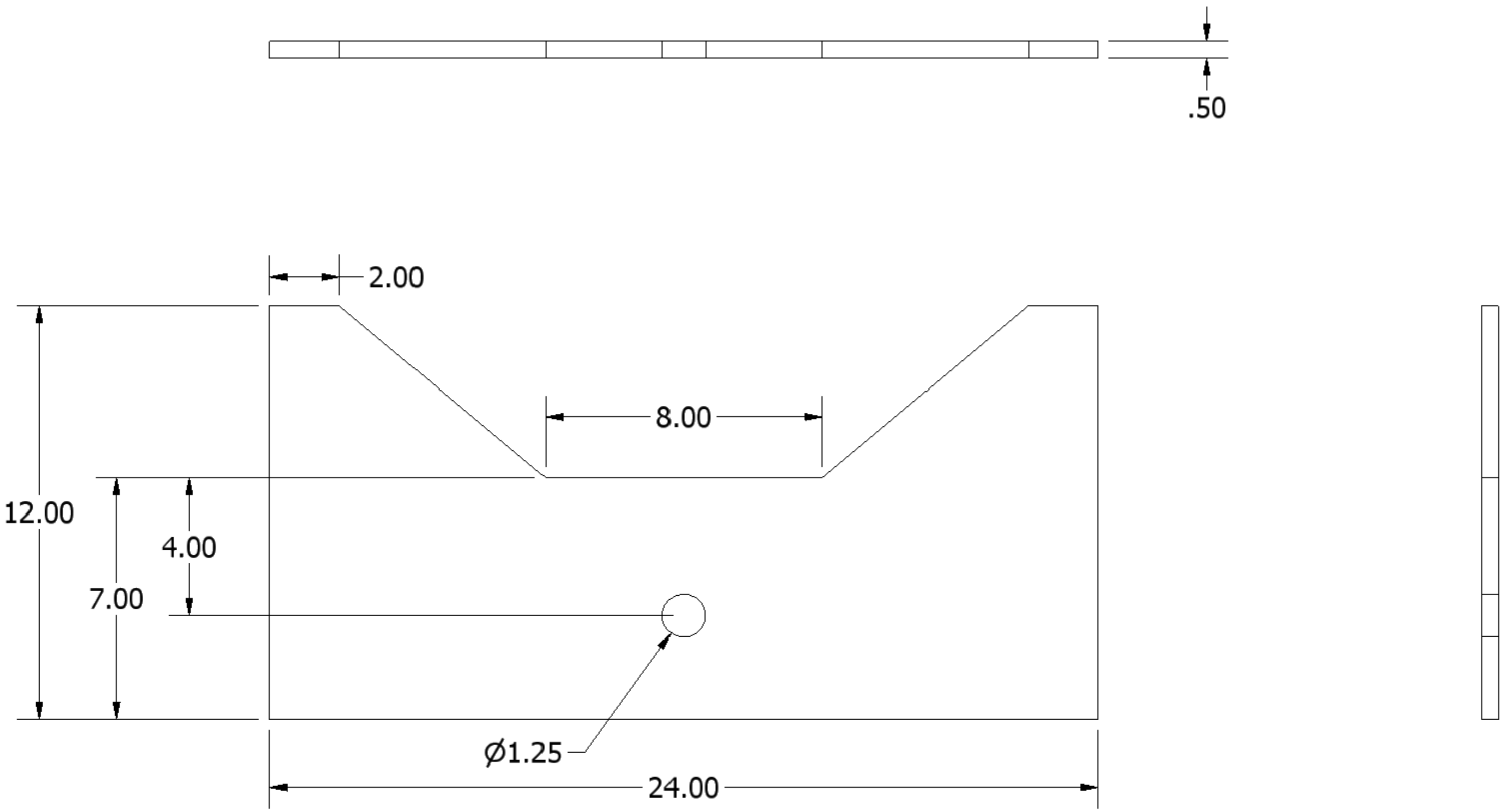
DRAWN: CHAD BROERING		INJURY BIOMECHANICS RESEARCH LAB
MATERIAL: A-36 HRS, OTHERS		TITLE: VERTICALTEST FRAME
ASSEMBLY: SUPPORT STRUCTURE		DESCRIPTION: VERTICAL MOTION SUPPORT STRUCTURE
DATE: 10/31/2008		
SHEET SIZE: A	SCALE	NOTES: SPECIFIC DIMENSIONS DETERMINED PER VENDOR
CHECKED BY:	1:20	SHEET 5 OF 26



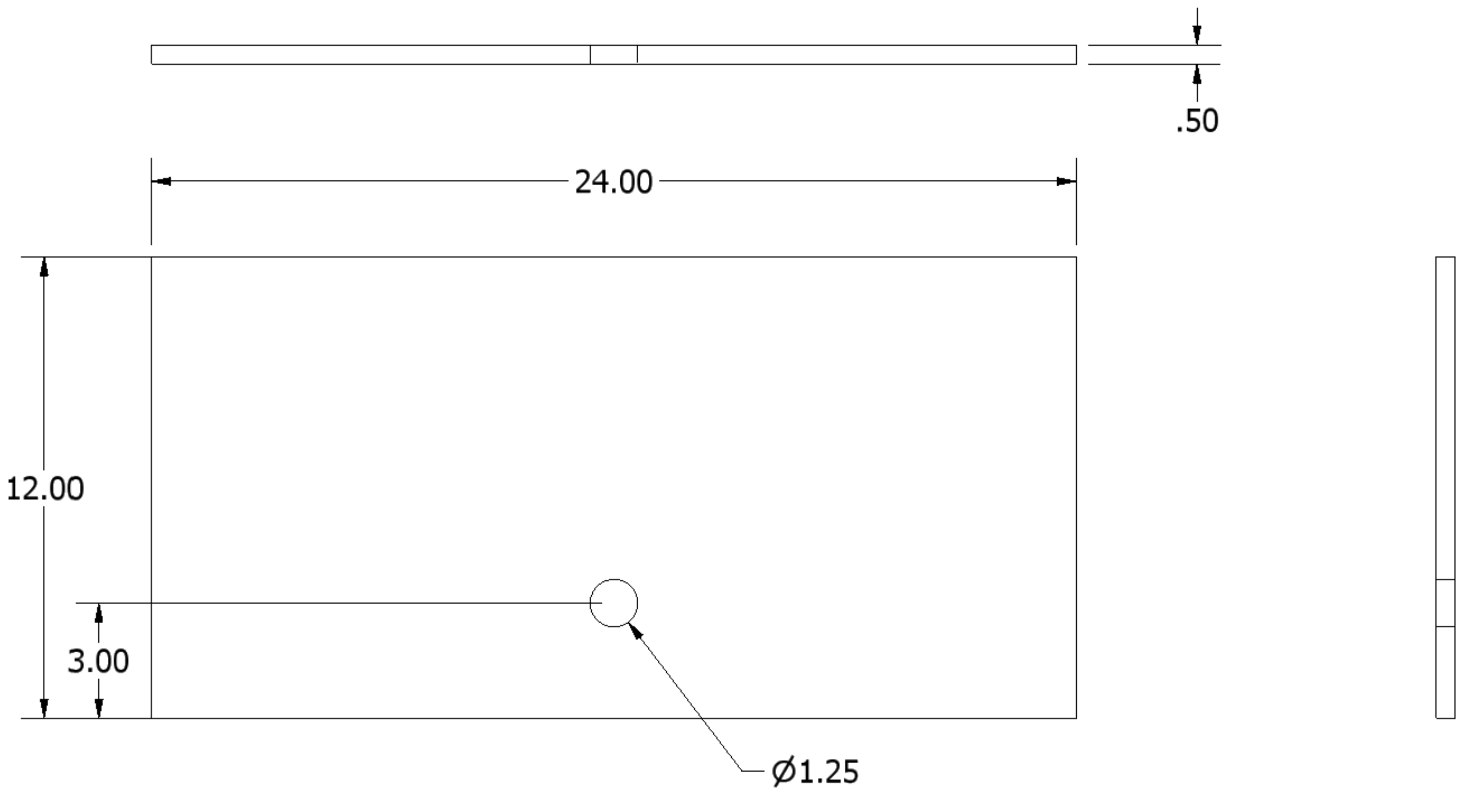
DRAWN: CHAD BROERING		INJURY BIOMECHANICS RESEARCH LAB	
MATERIAL: A-36 HRS		TITLE: CARRIAGE PLATE	
ASSEMBLY: CARRIAGE ASSEMBLY		DESCRIPTION: SIDES PLATE FOR CARRIAGE ASSEMBLY	
DATE: 10/31/2008			
SHEET SIZE: A	SCALE	NOTES: HOLE LOCATIONS NOT SPECIFIED.	
CHECKED BY:	1:8		SHEET 6 OF 26



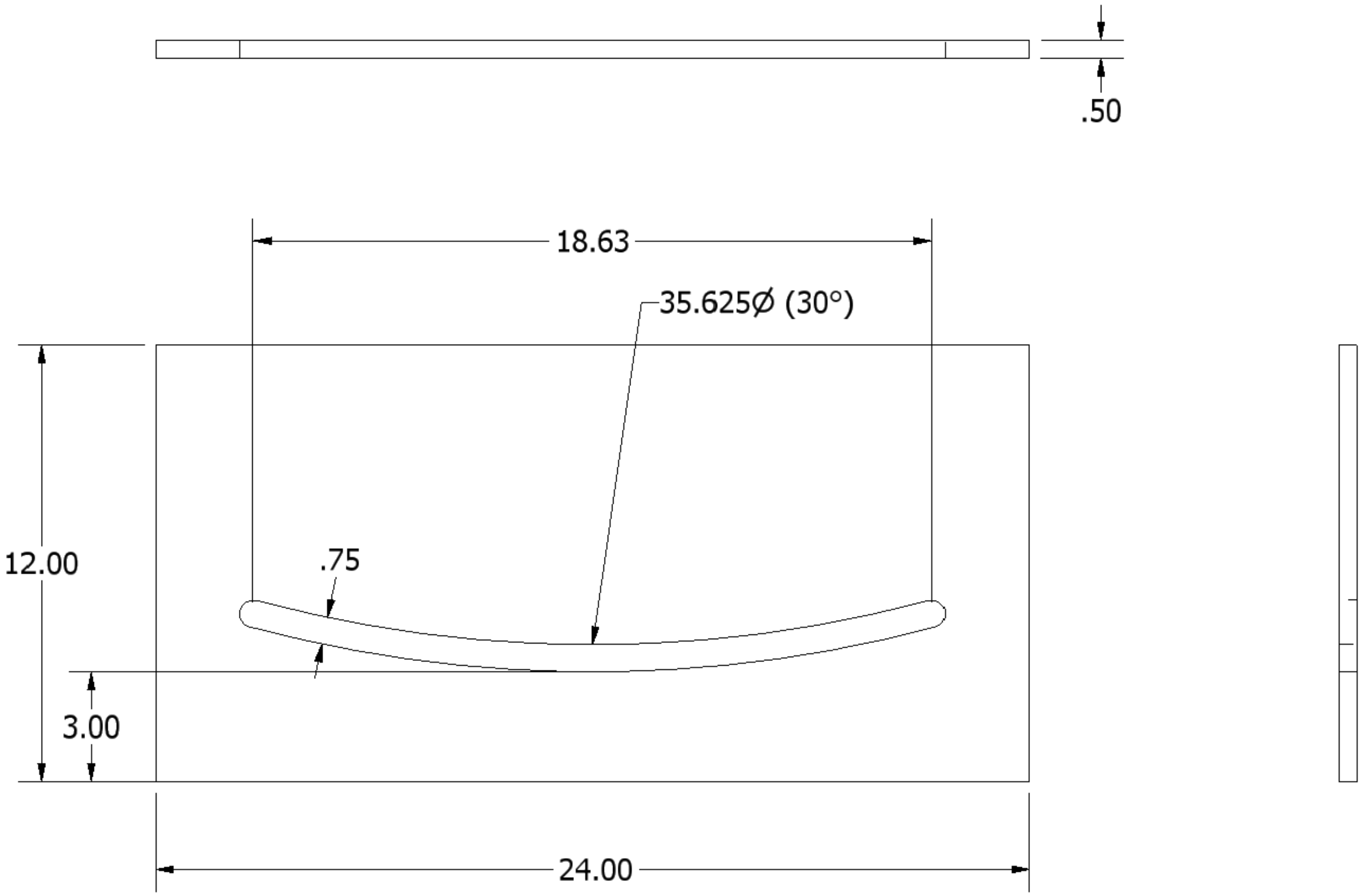
DRAWN: CHAD BROERING		INJURY BIOMECHANICS RESEARCH LAB
MATERIAL: A-36 HRS		TITLE: CARRIAGE PLATE ANGLE IRON
ASSEMBLY: CARRIAGE ASSEMBLY		DESCRIPTION: CARRIAGE PLATES ANGLE IRON SIZE AND LOCATION SPECIFIED
DATE: 10/31/2008		
SHEET SIZE: A	SCALE	NOTES: HOLE LOCATIONS NOT SHOWN
CHECKED BY:	1:10	SHEET 7 OF 26



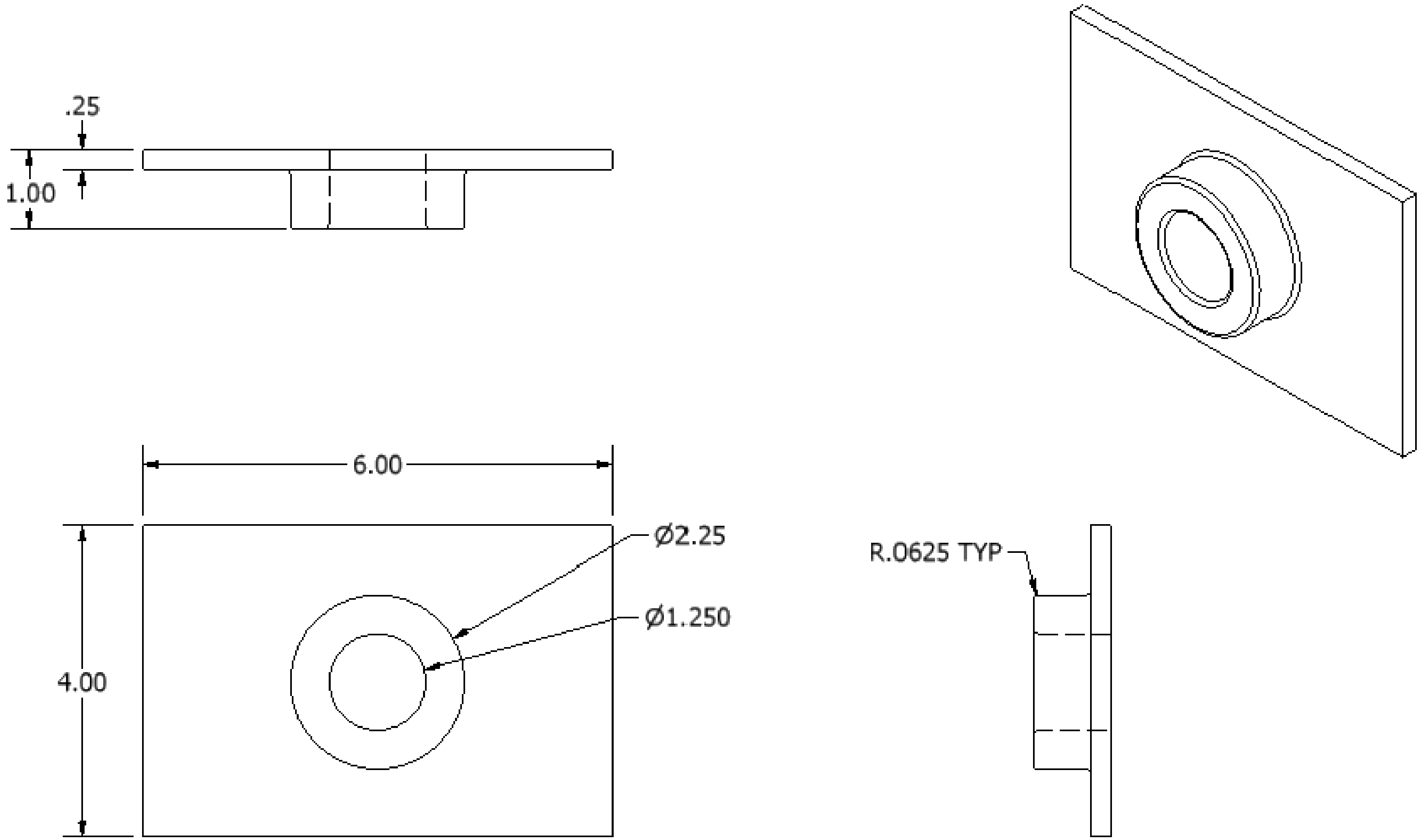
DRAWN: CHAD BROERING	INJURY BIOMECHANICS RESEARCH LAB
MATERIAL: A-36 HRS	TITLE: CARRIAGE TOP PLATE
ASSEMBLY: CARRIAGE ASSEMBLY	DESCRIPTION: PLATE LOCATED AT TOP-FRONT OF CARRIAGE ASSEMBLY
DATE: 10/31/2008	
SHEET SIZE: A / SCALE 1:5	NOTES: HOLE LOCATIONS NOT SHOWN
CHECKED BY:	SHEET 8 OF 26



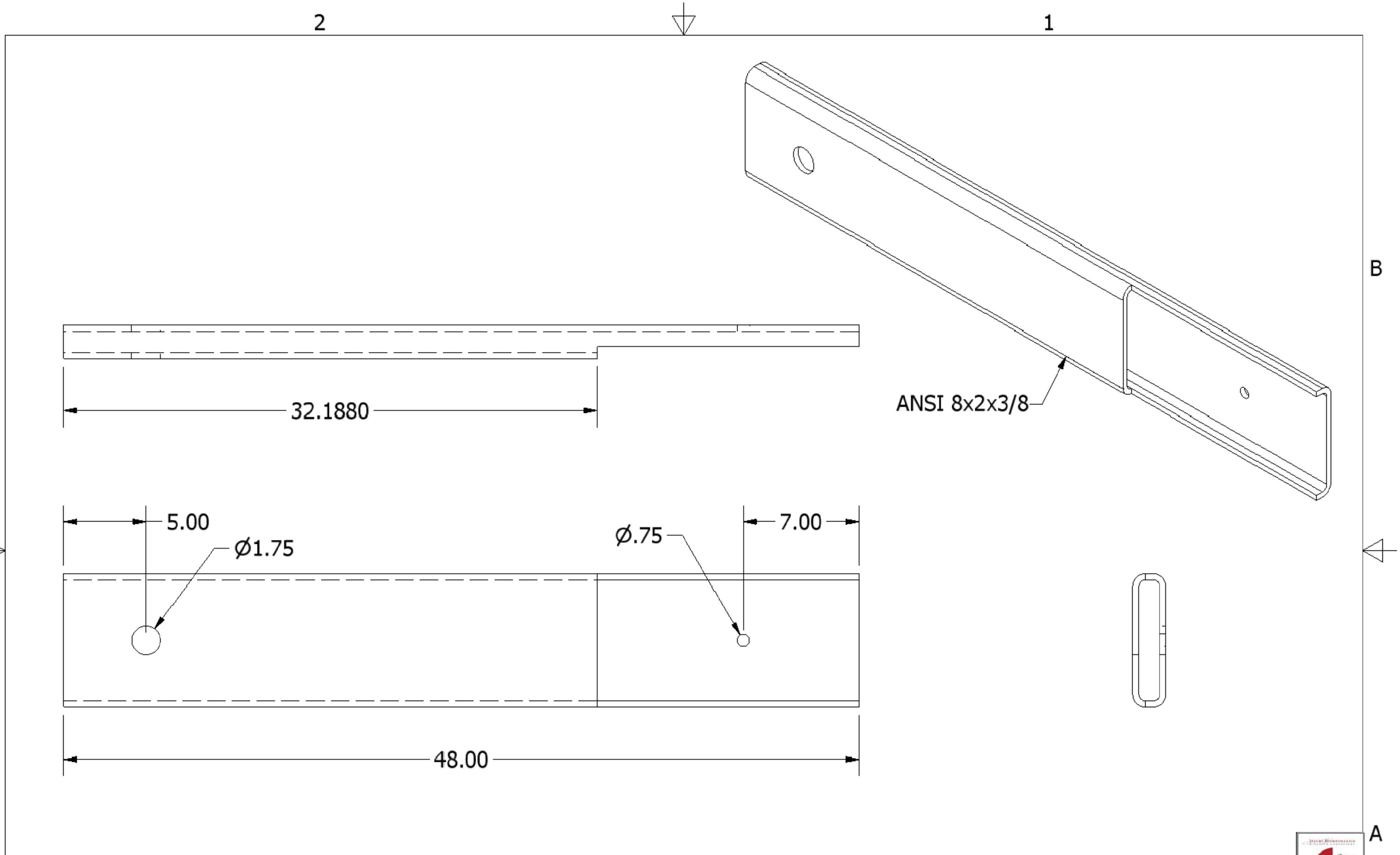
DRAWN: CHAD BROERING		INJURY BIOMECHANICS RESEARCH LAB	
MATERIAL: A-36 HRS		TITLE: CARRIAGE BOTTOM PLATE-FRONT	
ASSEMBLY: CARRIAGE ASSEMBLY		DESCRIPTION: PLATE LOCATED AT BOTTOM-FRONT OF CARRIAGE ASSEMBLY	
DATE: 10/31/2008			
SHEET SIZE: A	SCALE	NOTES: HOLE LOCATIONS NOT SHOWN	
CHECKED BY:	1:5		SHEET 9 OF 26



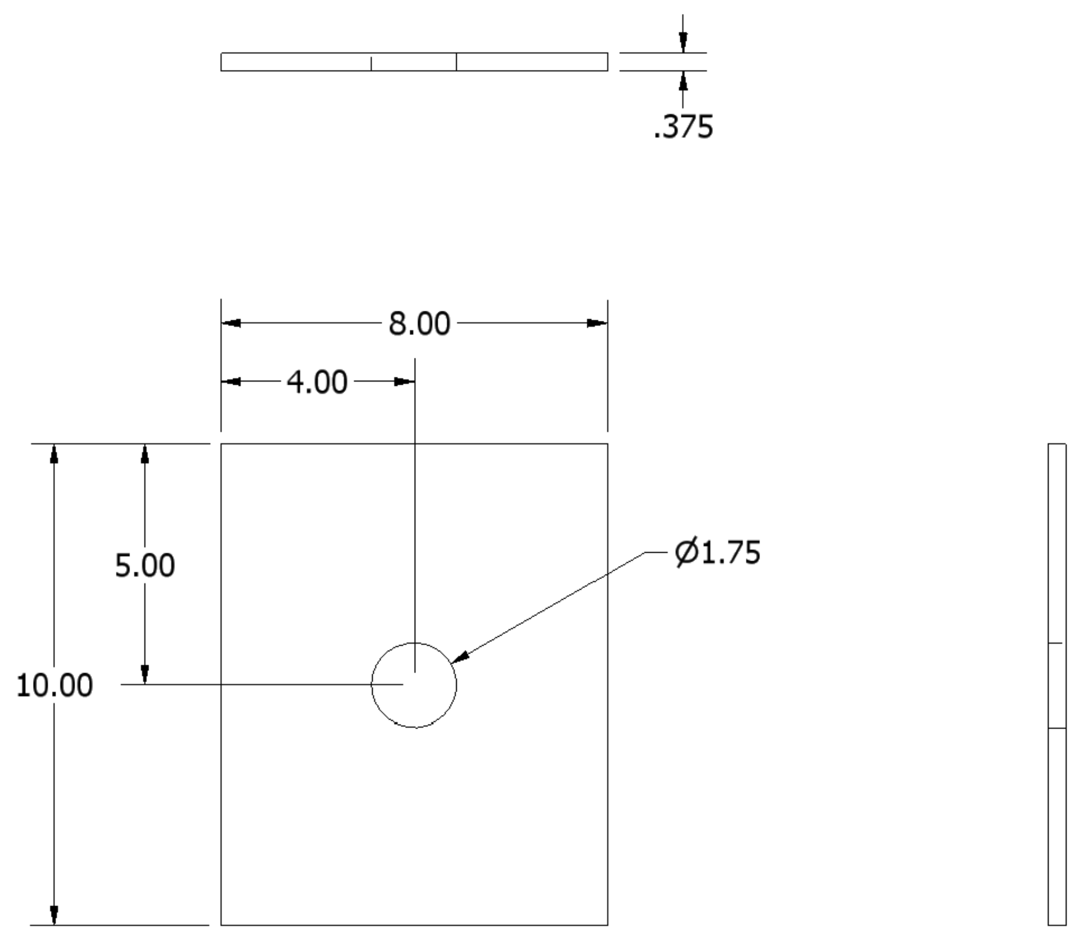
DRAWN: CHAD BROERING		INJURY BIOMECHANICS RESEARCH LAB	
MATERIAL: A-36 HRS		TITLE: CARRIAGE BOTTOM PLATE-REAR	
ASSEMBLY: CARRIAGE ASSEMBLY		DESCRIPTION: PLATE LOCATED AT BOTTOM-REAR OF CARRIAGE ASSEMBLY	
DATE: 10/31/2008			
SHEET SIZE: A	SCALE	NOTES: HOLE LOCATIONS NOT SHOWN	
CHECKED BY:	1:5		SHEET 10 OF 26



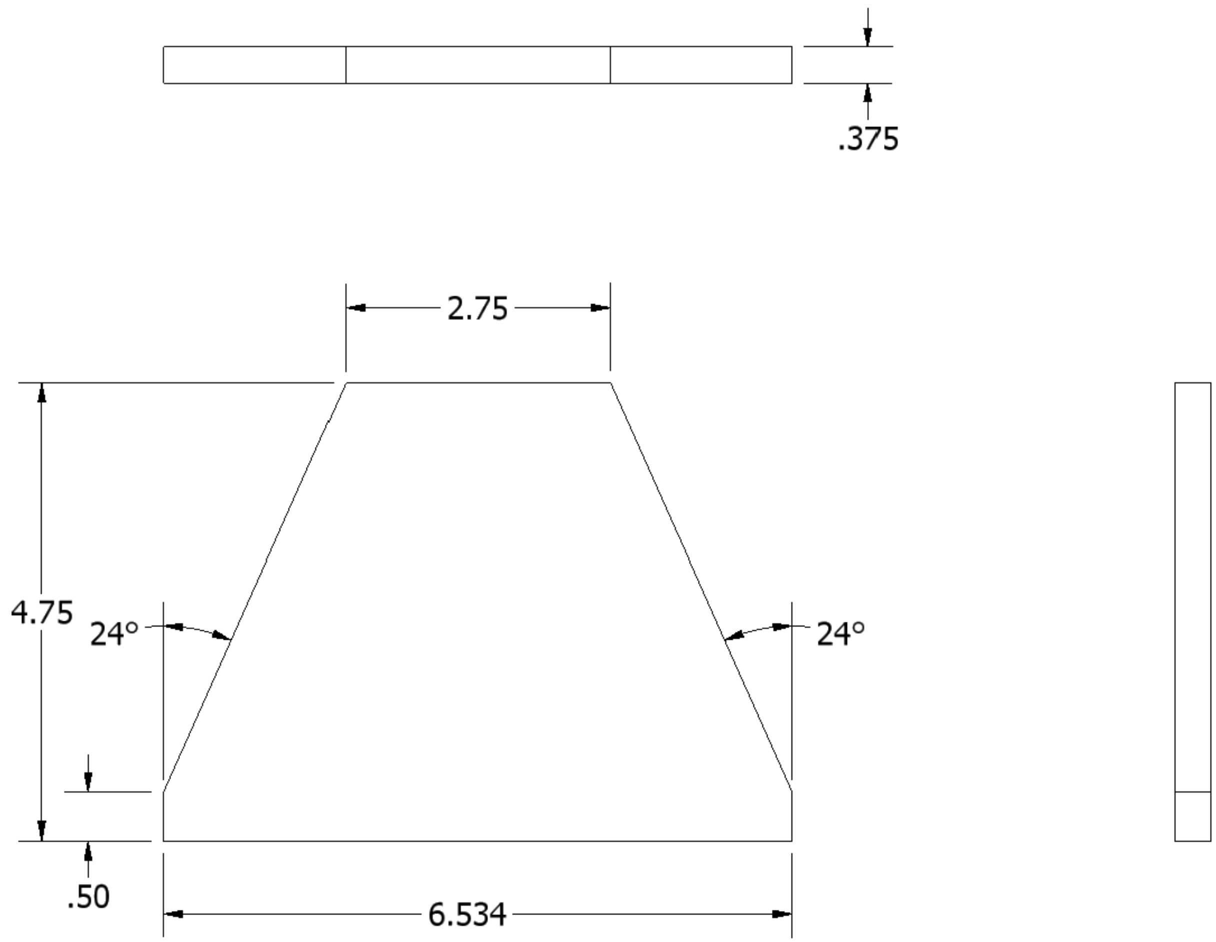
DRAWN: CHAD BROERING		INJURY BIOMECHANICS RESEARCH LAB	
MATERIAL: HRS		TITLE: CARRIAGE CLEVIS PIN MOUNTS	
ASSEMBLY: CARRIAGE ASSEMBLY		DESCRIPTION: SHAFT MOUNTS LOCATED AT TOP/BOTTOM OF FRONT PLATES	
DATE: 10/31/2008			
SHEET SIZE: A	SCALE	NOTES: HOLE LOCATIONS NOT SHOWN	
CHECKED BY:	1:2		SHEET 11 OF 26



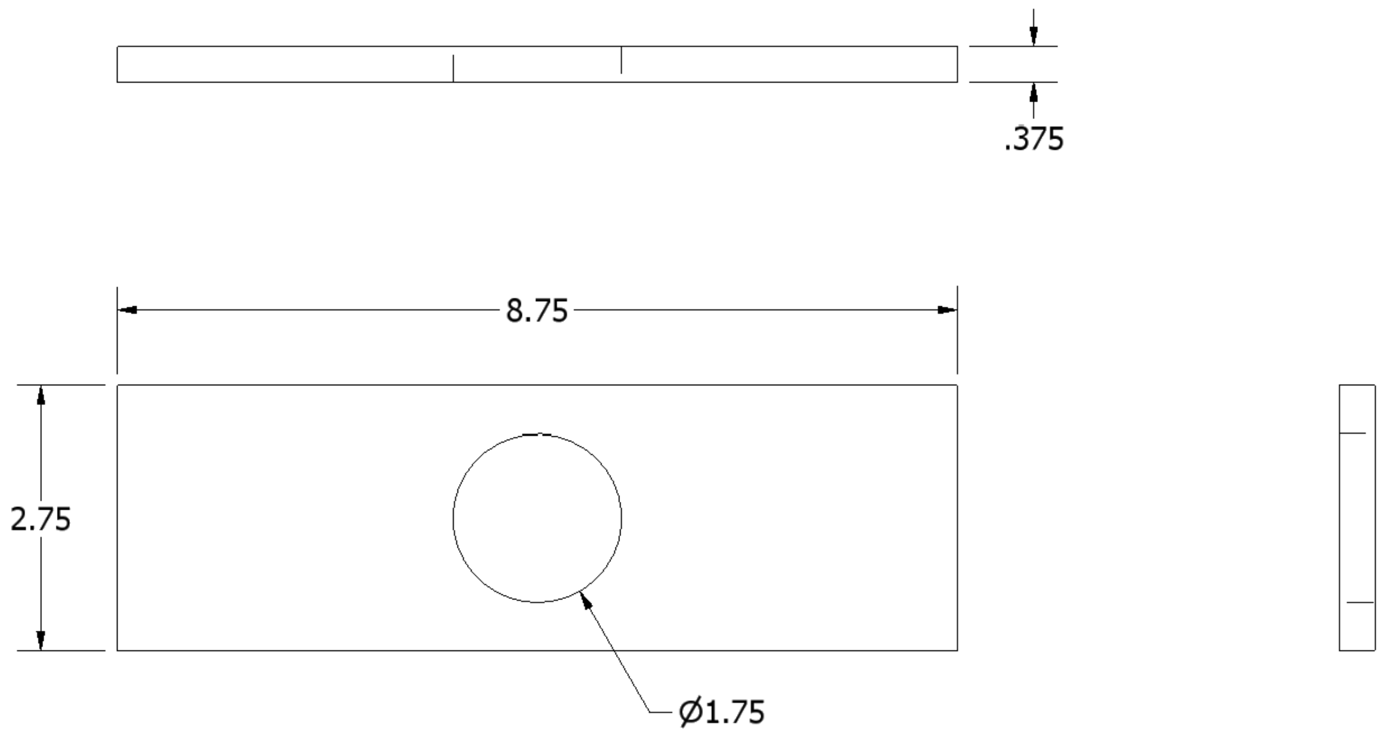
DRAWN: CHAD BROERING		INJURY BIOMECHANICS RESEARCH LAB	
MATERIAL: ASTM A-500 STEEL		TITLE: RECTANGULAR SUPPORT TUBE	
ASSEMBLY: CARRIAGE ASSEMBLY		DESCRIPTION: RECTANGULAR TUBE MOUNT FOR ACCELERATOR AND RAM ASSEMBLY	
DATE: 10/31/2008			
SHEET SIZE: A	SCALE	NOTES: HOLE LOCATIONS NOT SHOWN	
CHECKED BY:	1:8		SHEET 12 OF 26



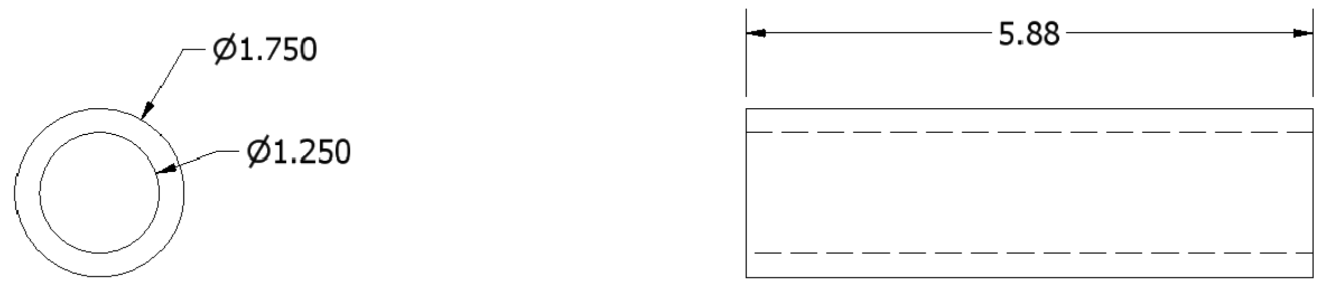
DRAWN: CHAD BROERING		INJURY BIOMECHANICS RESEARCH LAB	
MATERIAL: A-36 HRS		TITLE: BEARING MOUNTING PLATE	
ASSEMBLY: CARRIAGE ASSEMBLY		DESCRIPTION: BEARING MOUNTING PLATE (MOUNTED TO RECTANGULAR TUBING)	
DATE: 10/31/2008			
SHEET SIZE: A	SCALE	NOTES: WELD AND HOLE LOCATIONS NOT SHOWN	
CHECKED BY:	1:4		SHEET 13 OF 26



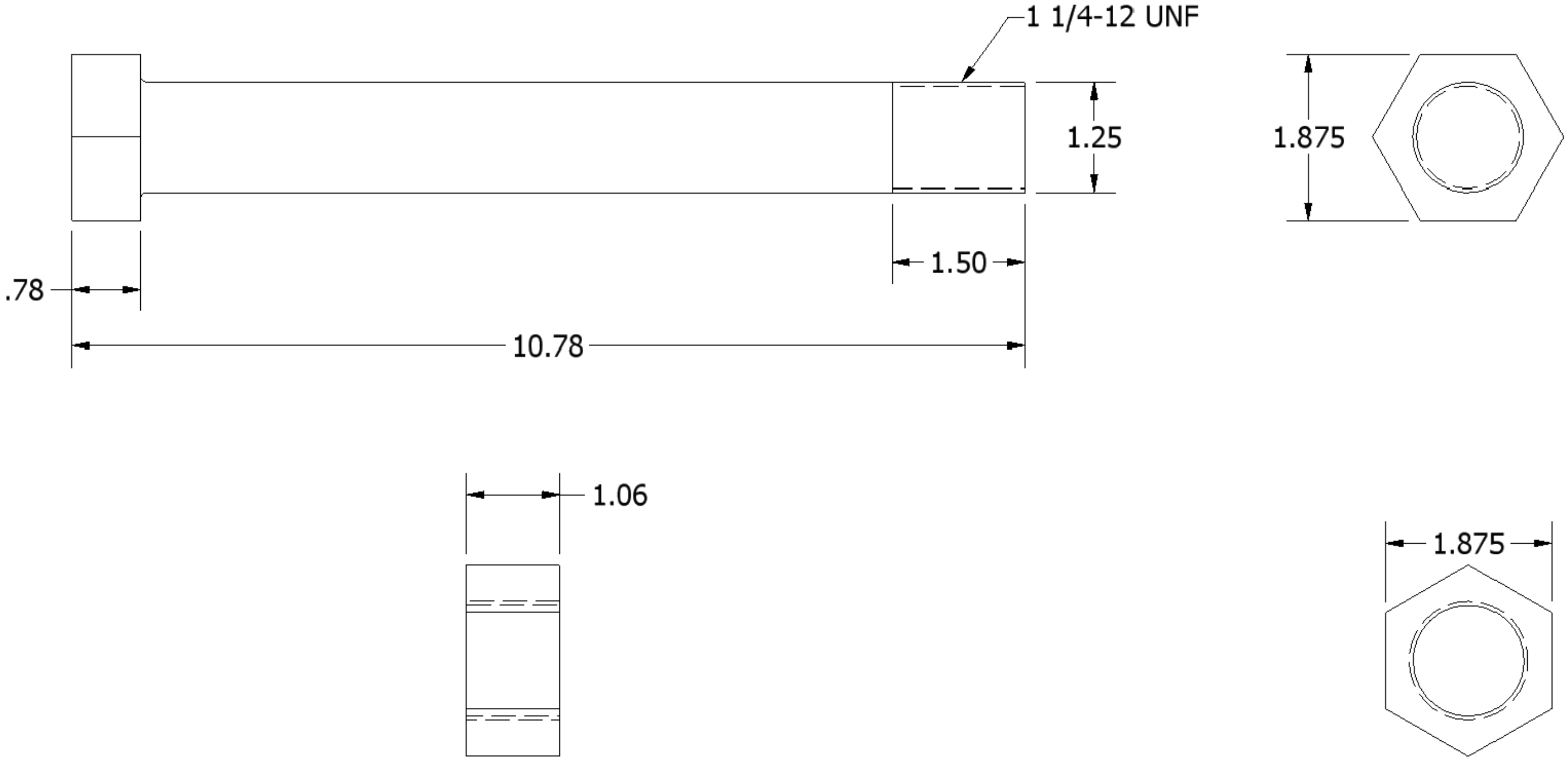
DRAWN: CHAD BROERING		INJURY BIOMECHANICS RESEARCH LAB
MATERIAL: A-36 HRS		TITLE: BEARING SIDE PLATE
ASSEMBLY: CARRIAGE ASSEMBLY		DESCRIPTION: SIDE PLATES (MOUNTED TO BEARING MOUNTING PLATE)
DATE: 10/31/2008		
SHEET SIZE: A	SCALE	NOTES: WELD AND HOLE LOCATIONS NOT SHOWN
CHECKED BY:	1:2	SHEET 14 OF 26



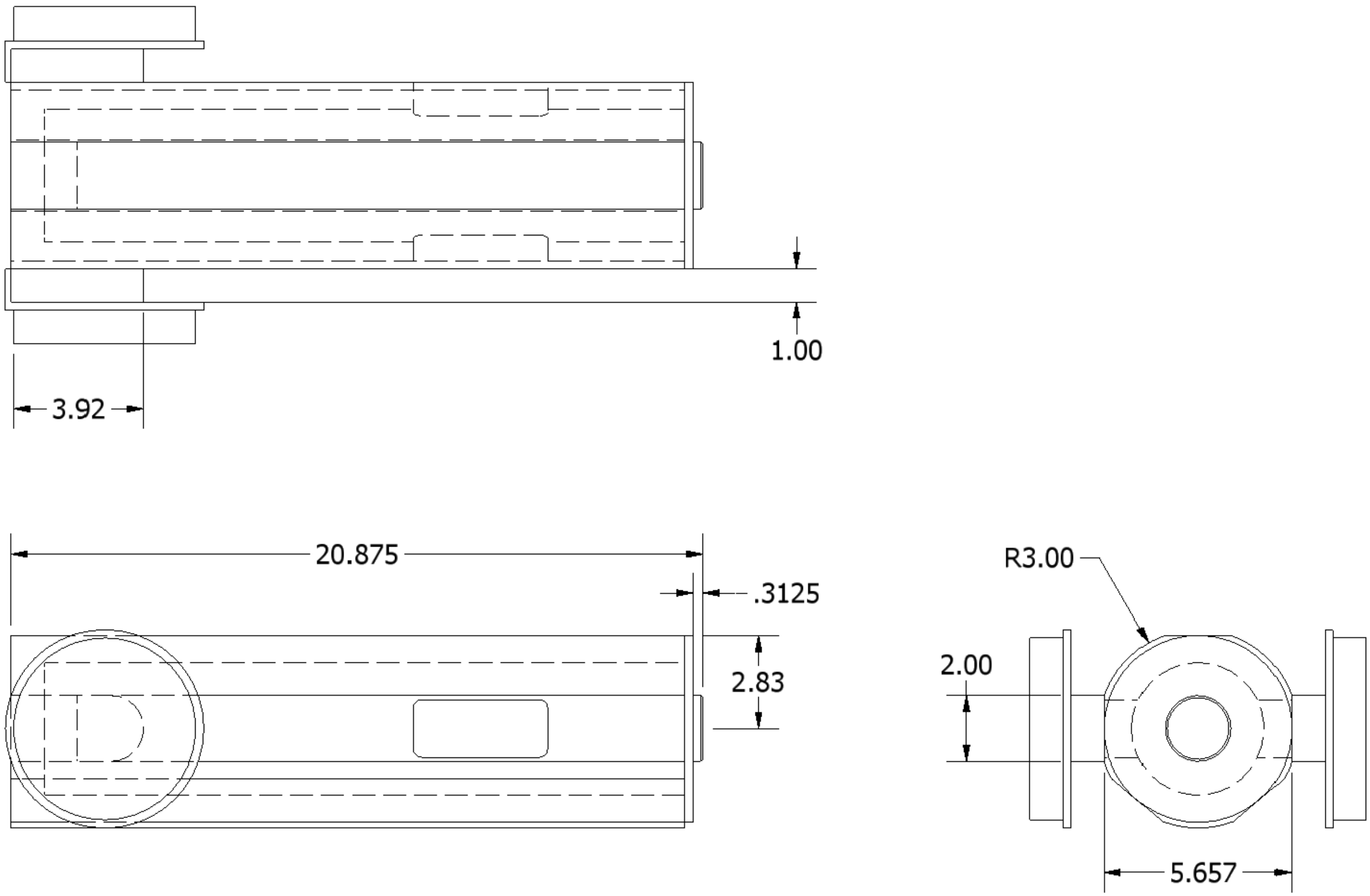
DRAWN: CHAD BROERING	INJURY BIOMECHANICS RESEARCH LAB
MATERIAL: A-36 HRS	TITLE: BEARING TOP PLATE
ASSEMBLY: CARRIAGE ASSEMBLY	DESCRIPTION: TOP PLATE (MOUNTED TO BEARING SIDE PLATES)
DATE: 10/31/2008	
SHEET SIZE: A — SCALE 1:2	NOTES: WELD AND HOLE LOCATIONS NOT SHOWN
CHECKED BY:	SHEET 15 OF 26



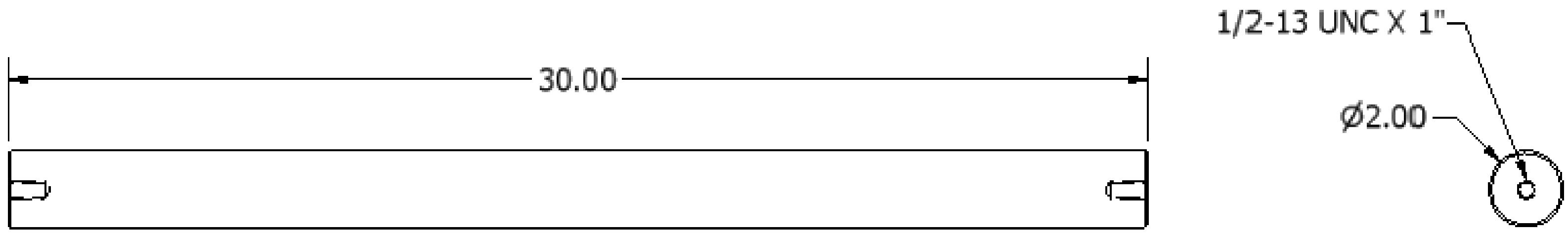
DRAWN: CHAD BROERING		INJURY BIOMECHANICS RESEARCH LAB	
MATERIAL: A-36 HRS		TITLE: CLEVIS PIN TUBE	
ASSEMBLY: CARRIAGE ASSEMBLY		DESCRIPTION: HOLLOW TUBE WELDED TO BEARING SUPPORT STRUCTURE (FOR CLEVIS PIN)	
DATE: 10/31/2008			
SHEET SIZE: A	SCALE	NOTES: WELD LOCATIONS NOT SHOWN	
CHECKED BY:	1:2		SHEET 16 OF 26



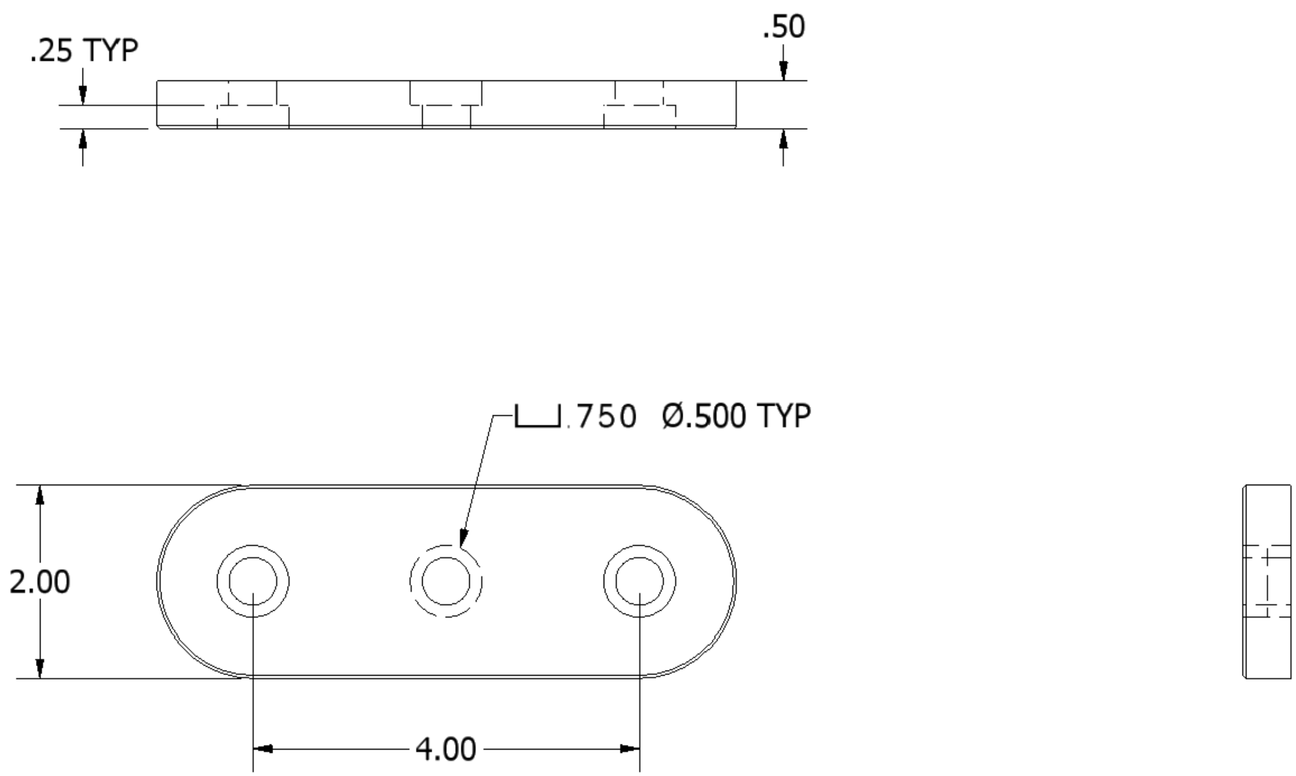
DRAWN: CHAD BROERING		INJURY BIOMECHANICS RESEARCH LAB	
MATERIAL: CRS		TITLE: CLEVIS PIN/NUT	
ASSEMBLY: CARRIAGE ASSEMBLY		DESCRIPTION: 1.25" CLEVIS PIN AND NUT FOR ROTATIONAL JOINT	
DATE: 10/31/2008			
SHEET SIZE: A	SCALE	NOTES: MOUNTED TO BEARING ASSEMBLY AND RECTANGULAR TUBING	
CHECKED BY:	1:2		SHEET 17 OF 26



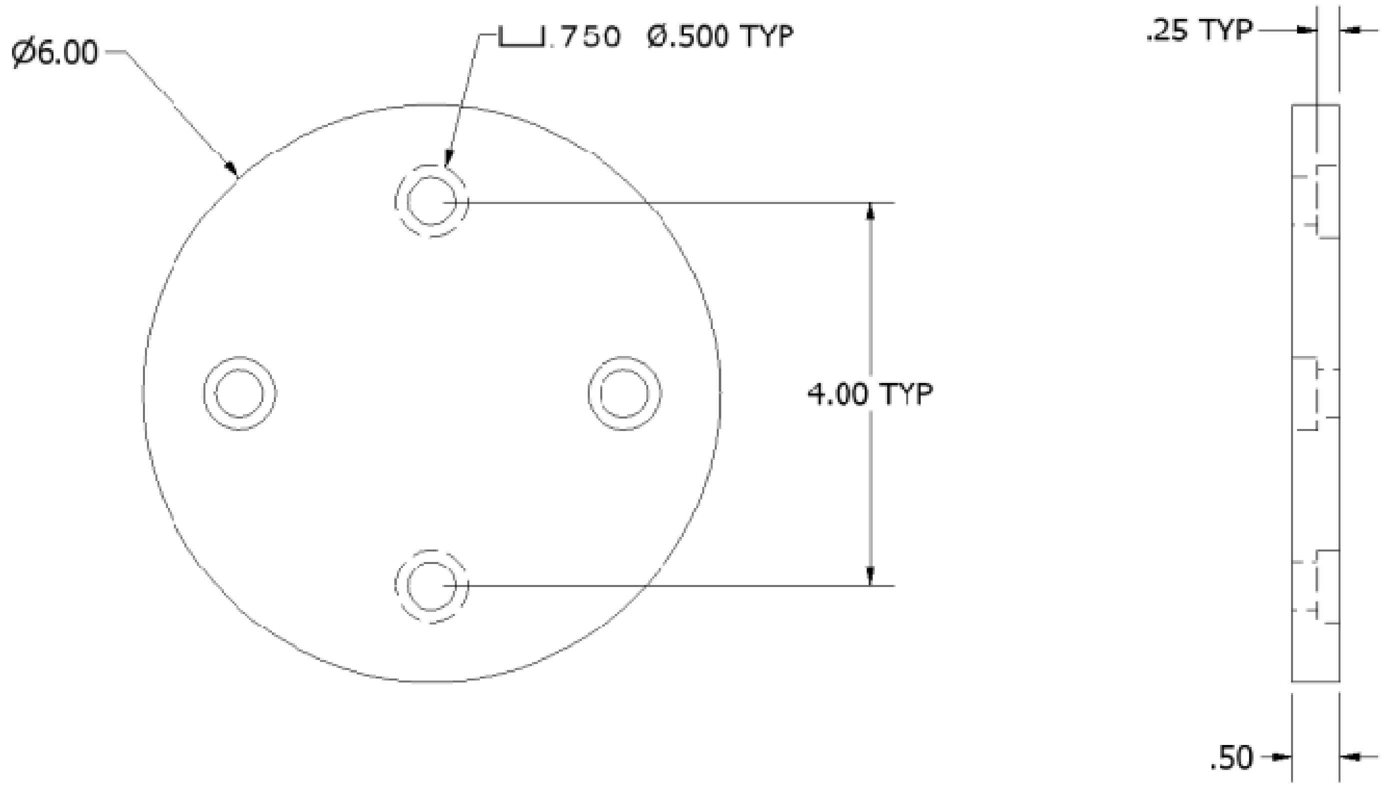
DRAWN: CHAD BROERING		INJURY BIOMECHANICS RESEARCH LAB	
MATERIAL: ALUMINUM		TITLE: PNEUMATIC ACCELERATOR	
ASSEMBLY: RAM ASSEMBLY		DESCRIPTION: ROUGH SKETCH OF CURRENT PNEUMATIC ACCELERATOR	
DATE: 10/31/2008			
SHEET SIZE: A	SCALE	NOTES: MEASUREMENTS ARE WITHIN APPROXIMATELY 1/8"	
CHECKED BY:	1:5		SHEET 18 OF 26



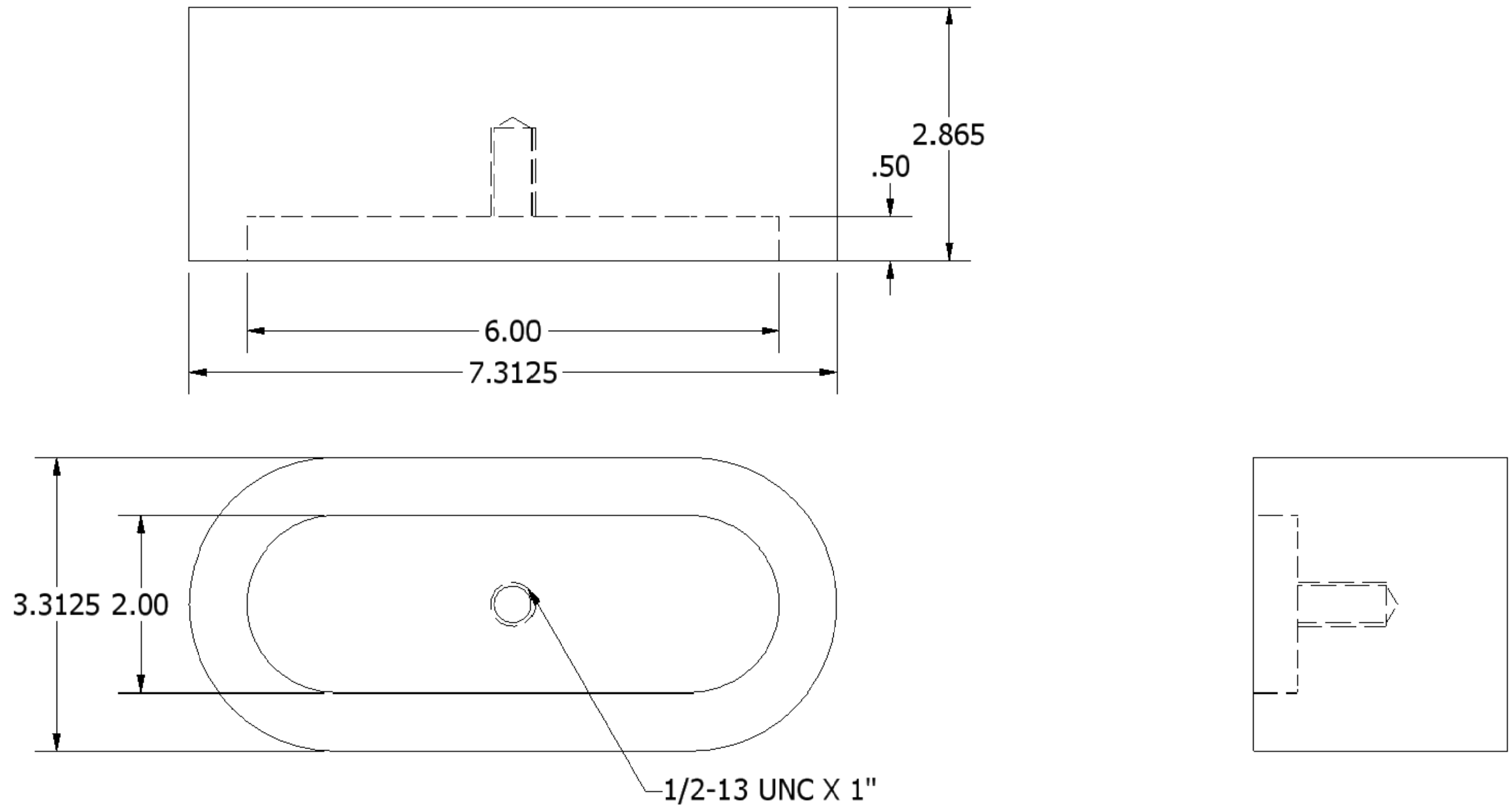
DRAWN: CHAD BROERING		INJURY BIOMECHANICS RESEARCH LAB	
MATERIAL: ALUMINUM		TITLE: RAM SHAFT	
ASSEMBLY: RAM ASSEMBLY		DESCRIPTION: RAM SHAFTS TO BE PURCHASED FROM SKAMAR	
DATE: 10/31/2008			
SHEET SIZE: A	SCALE	NOTES: CLASS "N" SPECIFICATIONS	
CHECKED BY:	1:5		SHEET 19 OF 26



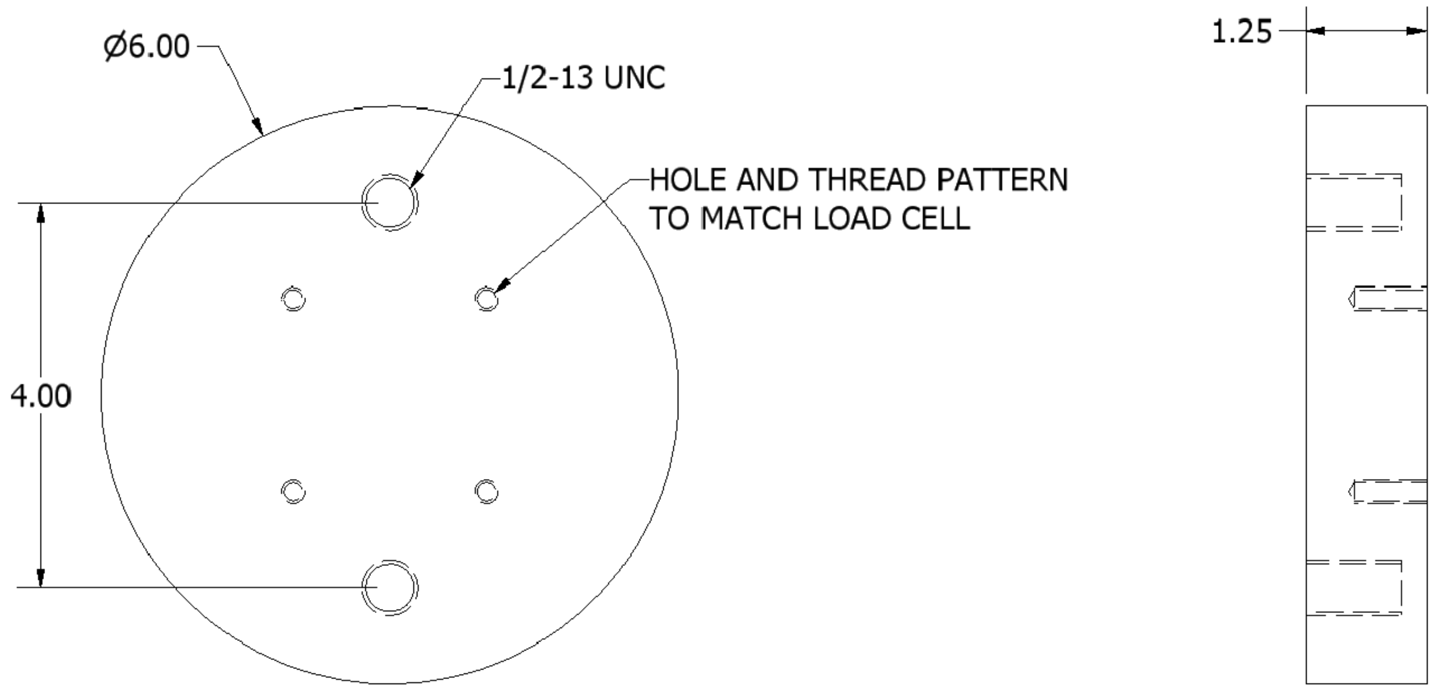
DRAWN: CHAD BROERING		INJURY BIOMECHANICS RESEARCH LAB	
MATERIAL: ALUMINUM		TITLE: RAM REAR PLATE	
ASSEMBLY: RAM ASSEMBLY		DESCRIPTION: REAR PLATE OF IMPACT RAM	
DATE: 10/31/2008			
SHEET SIZE: A	SCALE	NOTES:	
CHECKED BY:	1:2		SHEET 20 OF 26



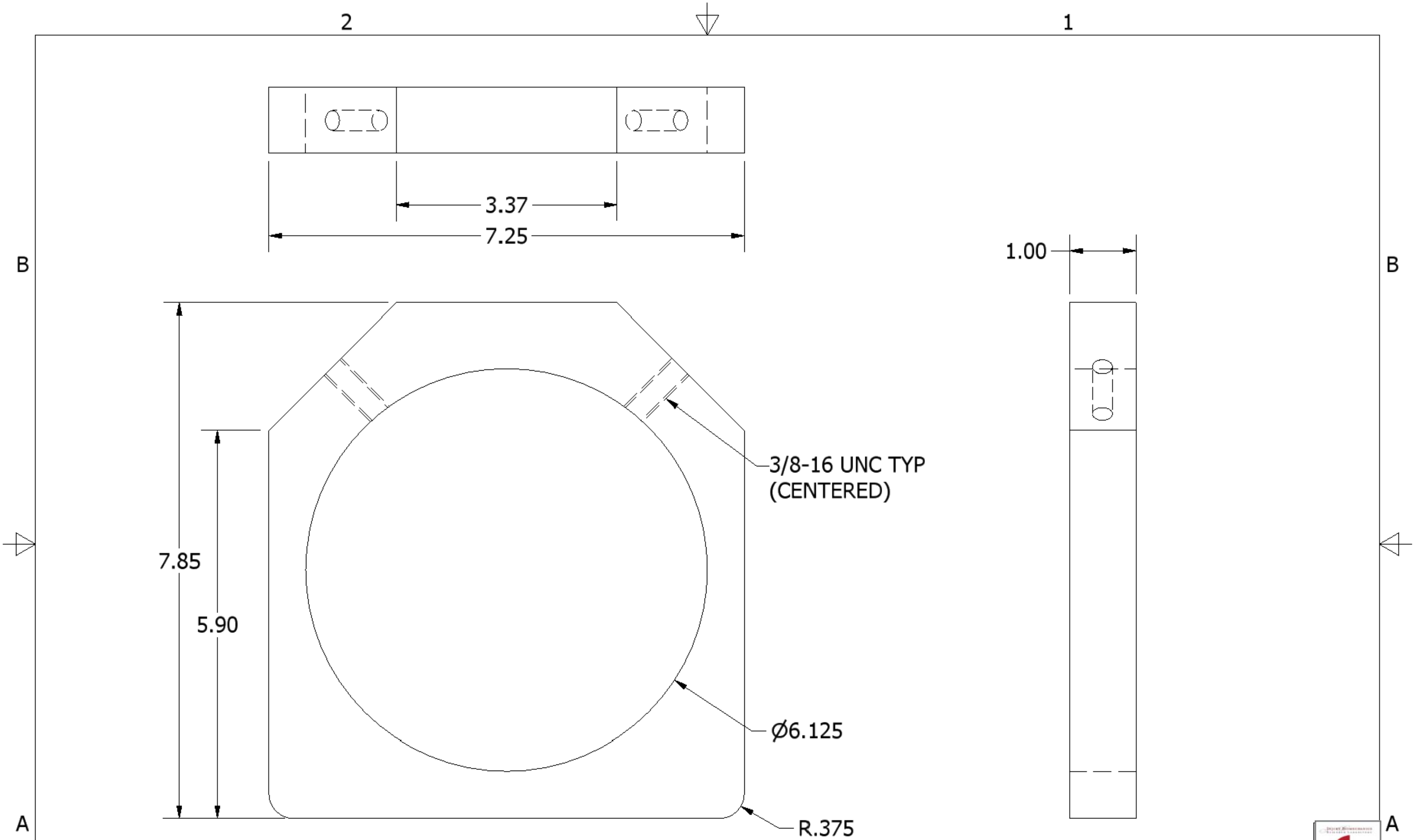
DRAWN: CHAD BROERING		INJURY BIOMECHANICS RESEARCH LAB	
MATERIAL: ALUMINUM		TITLE: RAM FRONT PLATE	
ASSEMBLY: RAM ASSEMBLY		DESCRIPTION: FRONT PLATE OF IMPACT RAM	
DATE: 10/31/2008			
SHEET SIZE: A	SCALE	NOTES:	
CHECKED BY:	1:2		SHEET 21 OF 26



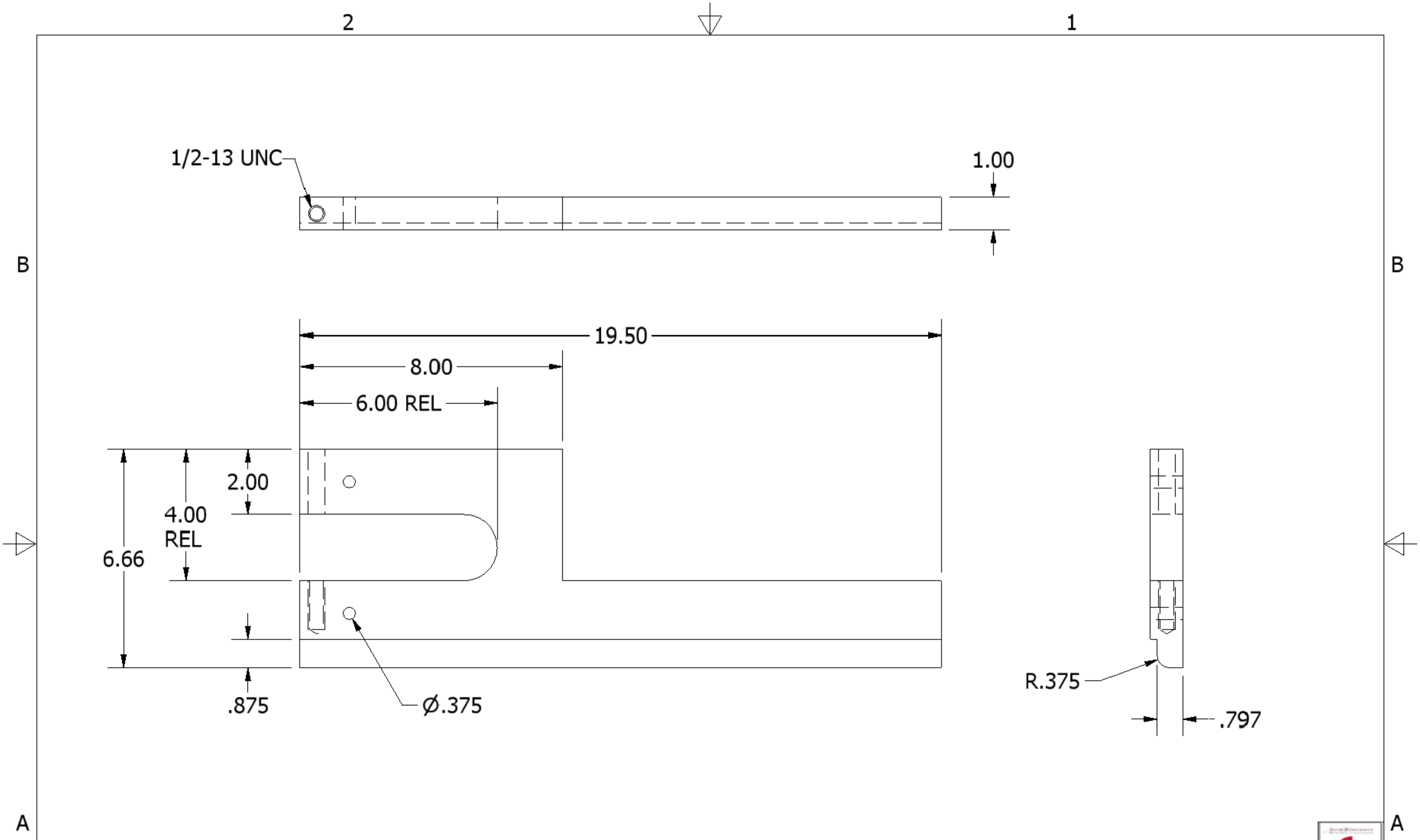
DRAWN: CHAD BROERING		INJURY BIOMECHANICS RESEARCH LAB
MATERIAL: HRS		TITLE: RAM REAR WEIGHT
ASSEMBLY: RAM ASSEMBLY		DESCRIPTION: REAR WEIGHT OF IMPACT RAM
DATE: 10/31/2008		
SHEET SIZE: A	SCALE	NOTES: DIMENSIONS ARE CHOSEN TO ACHIEVE PREDETERMINED WEIGHT
CHECKED BY:	1:2	SHEET 22 OF 26



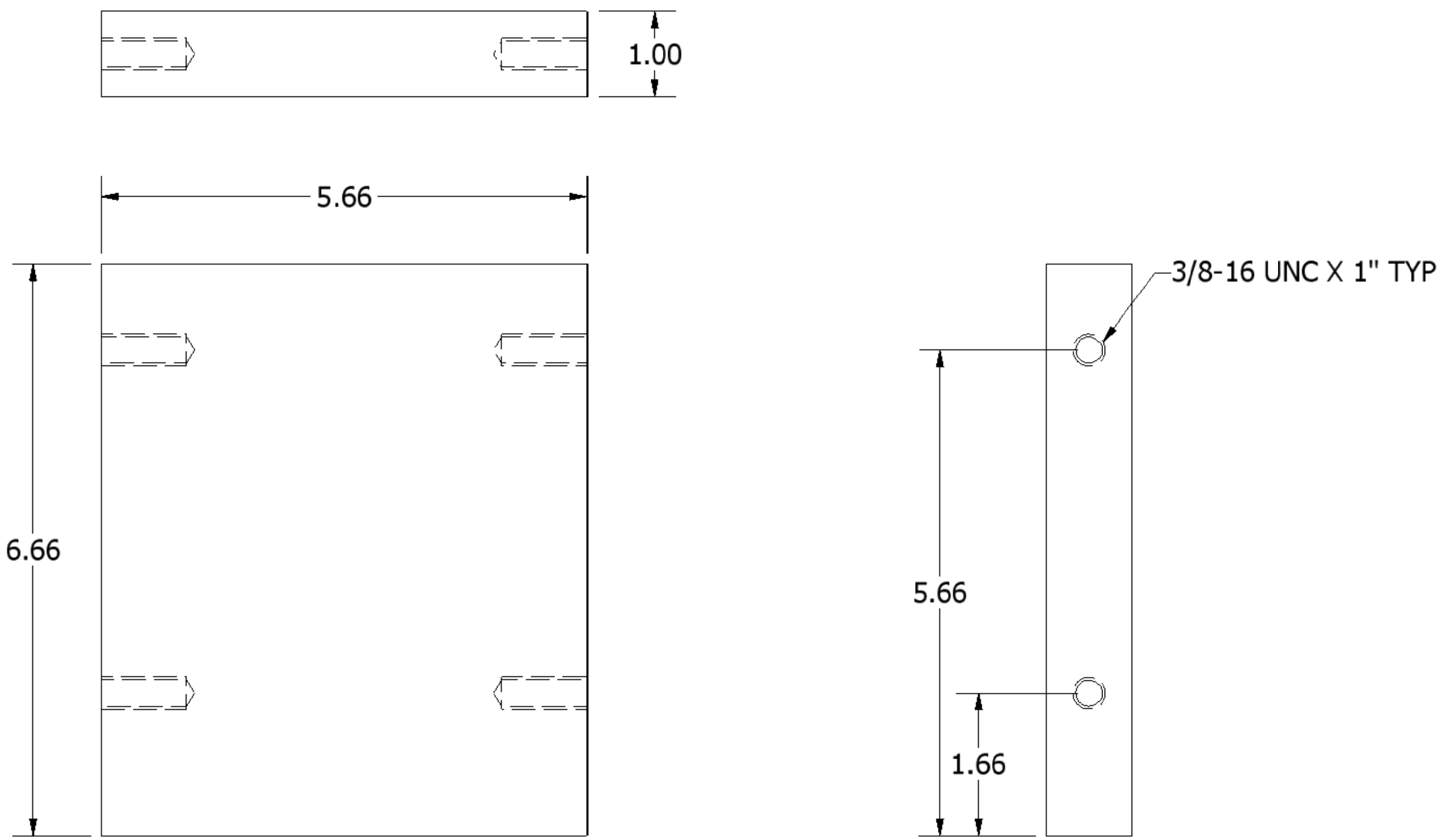
DRAWN: CHAD BROERING		INJURY BIOMECHANICS RESEARCH LAB	
MATERIAL: HRS		TITLE: RAM FRONT WEIGHT	
ASSEMBLY: RAM ASSEMBLY		DESCRIPTION: FRONT WEIGHT OF IMPACT RAM	
DATE: 10/31/2008			
SHEET SIZE: A	SCALE	NOTES: DIMENSIONS ARE CHOSEN TO ACHIEVE PREDETERMINED WEIGHT	
CHECKED BY:	1:2		SHEET 23 OF 26



DRAWN: CHAD BROERING		INJURY BIOMECHANICS RESEARCH LAB
MATERIAL: HRS		TITLE: ACCELERATOR MOUNTING BRACKET-FRONT
ASSEMBLY: RAM ASSEMBLY		DESCRIPTION: MOUNTING HOLD DOWN BRACKET FOR FRONT OF ACCELERATOR
DATE: 10/31/2008		
SHEET SIZE: A	SCALE	NOTES: THREADS FOR LOCATING PINS - MOUNTING LOCATIONS NOT SHOWN
CHECKED BY:	1:2	SHEET 24 OF 26



DRAWN: CHAD BROERING		INJURY BIOMECHANICS RESEARCH LAB	
MATERIAL: HRS		TITLE: ACCELERATOR MOUNTING BRACKET-SIDE	
ASSEMBLY: RAM ASSEMBLY		DESCRIPTION: MOUNTING BRACKET FOR SIDE OF ACCELERATOR	
DATE: 10/31/2008			
SHEET SIZE: A	SCALE	NOTES: HOLE LOC'NS NOT SHOWN, REL = RELATIVE TO ACCELERATOR SPECS	
CHECKED BY:	1:4		SHEET 25 OF 26



DRAWN: CHAD BROERING		INJURY BIOMECHANICS RESEARCH LAB	
MATERIAL: HRS		TITLE: ACCELERATOR MOUNTING BRACKET-REAR	
ASSEMBLY: RAM ASSEMBLY		DESCRIPTION: MOUNTING BRACKET FOR REAR OF ACCELERATOR	
DATE: 10/31/2008			
SHEET SIZE: A	SCALE	NOTES: HOLE LOC'NS NOT SHOWN, RELATIVE TO ACTUAL ACCELERAOR SPECS	
CHECKED BY:	1:2		SHEET 26 OF 26